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DESIGN SELECTION TESTS FOR TRAC RETRACTION MECHANISM

Sikorsky Aircraft Division
United Technologies Corporation
Stratford, Conn. 06602

January 1977

Final Report

Approved for public release;
distribution unlimited.

Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION PAPER

This report has been reviewed by the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory and is considered to be technically sound. The program was initiated to perform design selection tests of the telescoping rotor (TRAC) system mechanism, using full-scale components to determine the adequacy and reliability of the full-scale retraction mechanism.

Mr. Patrick A. Cancro of the Aeromechanics Technical Area, Technology Applications Division, served as project engineer for this effort.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

An investigation was conducted to select appropriate materials for and to demonstrate the satisfactory operation of a full-scale jackscrew and nut assembly for a TRAC variable-diameter helicopter rotor blade.

The investigation successfully demonstrated the operation of a suitable material combination in the jackscrew/nut configuration,

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20. ABSTRACT (Cont'd)

cont with more than 300 full retraction-extension cycles under full design load and rotational speed conditions. The operation of the system improved significantly as the test progressed, with the coefficient of friction, the temperature rise, and the wear rate all decreasing with time. This result confirms that successful full-scale development of the TRAC rotor concept is achievable. It was concluded that a wear life on the order of 1000 retraction-extension cycles can be achieved in a production design.

A materials evaluation phase of the investigation utilized rings of selected material combinations to simulate a single thread contact. The preferred material/lubricant combination that emerged from these tests was a high-strength maraging-steel screw, coated with Vitrolube dry-film lubricant, and a carbon-graphite nut.

PREFACE

This investigation was conducted for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, under Contract DAAJ02-72-C-0049. Technical cognizance for the program was provided by Mr. Patrick A. Cancro and Mr. John W. Sobczak.

Mr. Harold K. Frint was the Sikorsky Task Manager for the overall direction of the program. Mr. Evan A. Fradenburgh provided technical assistance and direction, particularly with respect to blade design and load criteria. Mr. Donald Moyer of Pure Carbon Co. directed the materials evaluation test program. Messrs Edmond Conklin and Wayne Throp were responsible for the full-scale jackscrew/nut evaluation test program.

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INTRODUCTION

The concept of varying in-flight geometry to improve aerodynamic performance characteristics in specific flight modes has long been standard practice with fixed-wing aircraft. Some notable examples are the high-lift wing flaps, variable-pitch propellers, speed brakes, jet engine thrust reversers, variable-sweep wings, and retractable landing gear. Yet, except for the latter feature, helicopters generally have not used variable geometry devices for control of performance. The helicopter main rotor does incorporate collective and cyclic pitch controls but the primary purpose of this variable-geometry feature is to maintain trim attitude and provide directional control, rather than to modify performance characteristics.

Some of the improvements obtained as a result of in-flight variable geometry features in fixed-wing aircraft are: higher cruise speed, lower landing speed, longer range, greater maneuverability, reduced fuel consumption and smoother ride. These benefits are usually obtained at the expense of increased mechanical complexity and higher maintenance costs. However, as history will bear out, complex devices can be made to work reliably, and the resulting increased capability or improved economy permits survival in a hostile and competitive environment.

The Sikorsky Telescoping Rotor Aircraft (TRAC) is an in-flight variable-diameter rotor system designed to extend the capabilities and improve the performance of several categories of high-speed rotary-wing aircraft. Development of this rotor system began in 1966 at Sikorsky and has been supported by the Eustis Directorate of the U. S. Army Air Mobility Research and Development Laboratory since 1968.

Helicopters have by far the best hovering and low-speed flight characteristics of all VTOL aircraft configurations in terms of the power required and maneuverability. These desirable traits relate to the large diameter, low-disc-loading main rotor, which provides lift and control. At high cruise speeds, however, the conventional rotor system becomes more of a liability than an asset, particularly in the compound configuration. It is much less effective and efficient than the wing and control system of an airplane. In fact, one of the chief limitations to the wider utilization of rotary-wing aircraft is their rather low forward speed capability. Pure helicopters, for example, are limited to less than 200 knots, and conventional-rotor compound helicopters are limited to about 250 knots cruise speed.

The TRAC rotor is an attempt to combine the benefits of the helicopter at low speeds with the advantages of a conventional airplane in cruise flight. It accomplishes this by extending the rotor fully in hover and low-speed flight, and retracting the rotor to a smaller diameter in cruise, which reduces drag because of the reduced blade area and reduced tip speed, and also greatly reduces flapping sensitivity to gusts. Cruise speeds of up to approximately 350 knots can be obtained for the TRAC compound helicopter. For still higher cruise speeds, the TRAC rotor can be stopped in flight and folded away, providing a virtually unlimited speed potential.

Wind tunnel tests of a 9-foot-diameter aeroelastically-scaled rotor model have demonstrated the feasibility and some of the benefits of the TRAC concept. This test program, reported in detail in Reference 1, achieved all major objectives. The rotor was demonstrated successfully in every planned operating regime. Feasibility was established for both the high-speed compound helicopter and the stowed rotor aircraft configurations. Diameter changes were demonstrated at true forward speeds up to 150 knots at full rotational speed, rotor stops and starts at minimum diameter were demonstrated at true forward speeds up to 150 knots, and tests were conducted in the high-speed compound helicopter mode at true forward speeds up to 400 knots. The tests demonstrated many important benefits of the variable diameter capability, including improved performance and reduced vibration, stresses, and gust response. Other potential advantages of the TRAC rotor system have been reported in References 2 through 4.

---

[1] Fradenburgh, E. A., Murrill, R. J., and Kiely, E. F., DYNAMIC MODEL WIND TUNNEL TESTS OF A VARIABLE-DIAMETER, TELESCOPING-BLADE ROTOR SYSTEM (TRAC ROTOR), Sikorsky Aircraft Division, United Technologies Corporation; USAAMRDL Technical Report 73-32, Eustis Directorate, U. S. Army Air Mobility R&D Laboratory, Fort Eustis, Virginia, July 1973, AD771037.

[2] Segel, R. M., and Fradenburgh, E. A., DEVELOPMENT OF THE TRAC VARIABLE DIAMETER ROTOR CONCEPT; Paper presented at the AIAA/AHS VTOL Research, Design, and Operations Meeting, Georgia Institute of Technology, Atlanta, Georgia, February 17-19, 1969, AIAA Paper No. 69-221.

[3] Fradenburgh, E. A., EXTENSION OF COMPOUND HELICOPTER PERFORMANCE BY MEANS OF THE TELESCOPING ROTOR; Paper presented at the Air Force V/STOL Technology and Planning Conference, Las Vegas, Nevada, September 23-25, 1969.

[4] Fradenburgh, E. A., APPLICATION OF A VARIABLE DIAMETER ROTOR SYSTEM TO ADVANCED VTOL AIRCRAFT; Paper presented at the American Helicopter Society 31st National Forum, Washington, D. C., May 13-15, 1975.

Nearly all of the diameter-change system components on the TRAC blade, including the clutches and differential gears, represent routine mechanical system developments. The major exception to this is the jackscrew/nut combination. In order to keep system weight and retraction time to acceptably low values, the contact pressure, P, between screw and nut threads and the relative sliding velocity, V, must be relatively high. The combination of P and V (PV) falls outside the range of routine design practice. Another potential design problem is that a fairly large number of active threads are required to carry the large centrifugal force with a reasonably small and light screw. If all the threads were on a single nut, the elastic and thermal deformations in both screw and nut would make the thread loading distribution highly nonuniform. This problem is minimized by utilizing several independently-loaded nuts in the TRAC blade design.

The dynamic model blade tests, discussed above, operated at nominal contact pressures and sliding velocities that were both higher than those of the current full-scale preliminary design. This was very encouraging with regard to the probable success of a full-scale system, but not very much data on the friction and wear properties of the model jackscrew and nut assembly was obtained. Furthermore, it is known from heat transfer calculations that full-scale components, for a given PV value, develop higher surface contact temperatures than the model components, even though the total heat impulse per unit of mass, and therefore the mean temperature rise of the material, are the same. Thus, it was felt that a full-scale test of the jackscrew, with enough operational cycles to demonstrate wear properties and long-term friction behavior, was required.

The present investigation is intended to evaluate materials, lubricants, and problems associated with the fabrication, assembly, and operation of full-scale TRAC components. These include the outer blade spar, jackscrew, nuts, blade retention straps, torque tube, and bearing blocks. The scope of this program is to design and fabricate full-scale hardware and to test this in a specially designed facility that will permit blade retractions and extensions under simulated blade loads. The outer blade spar, torque tube, and bearing blocks were fabricated and selectively fitted, demonstrating that the dimensional tolerances needed to provide a proper sliding fit could be achieved.

Two full-scale jackscrew and mating nut assemblies were fabricated and tested. The materials selected for this initial design proved to be incompatible, necessitating a reevaluation of the material selection process. As a result, a laboratory materials research program was initiated from which potential jackscrew/nut configurations evolved. Tests were conducted to demonstrate acceptable friction and wear properties under repeated retraction and extension cycles with an applied axial loading simulating flight centrifugal loads.

FULL-SCALE DESIGN

GENERAL CONCEPT DESCRIPTION

A schematic arrangement of the TRAC rotor blade is shown in Figure 1. The basic actuation mechanism is a jackscrew that also serves as a primary tension member of the blade. Rotation of this screw imparts a linear retraction or extension motion to a series of retention nuts and, through tension straps, to the outboard half of the blade, which is the main lifting member. A torque tube, which is a streamlined ellipse in cross section, encloses the jackscrew, transmits blade pitch control motion to the outboard blade and carries bending moments across the sliding joint. When the rotor diameter is reduced, the outboard blade slides over and encloses the torque tube. The outboard blade, with a full airfoil cross section, comprises the outer half of the radius when the blade is extended. The blade planform is unusual in that the effective root cutout in the extended position is very large; however, even for a conventional blade, the outboard half typically produces 90 percent of the total lift in hover, and the large root cutout produces only a few percent loss in hover efficiency. The schematic drawing of Figure 1 does not show a number of important design features, including the multiple nut/strap assembly, buttress thread jackscrew structure, or sliding block arrangement. These items are discussed in detail in the following sections.

The TRAC blade system is based on a 56-foot diameter, four-bladed rotor. This size is ideal for full-scale testing in the 40-by 80-foot NASA-Ames Wind Tunnel and for flight demonstration on a Sikorsky S-61 helicopter or on the NASA/Army Rotor System Research Aircraft (RSRA) (see Figure 2). Although the standard S-61 rotor is 62 feet in diameter and has five blades rather than four, the wider chord of the TRAC blades gives the two rotors approximately equal useful blade areas and lifting capabilities.

Early in the conceptual design study, numerous schemes were evaluated for their potential in achieving a variable-diameter rotor system. These included the use of cables, rack-and-pinion devices, hydraulic and pneumatic systems, elastic schemes and jackscrews (both conventional and antifriction types). A conventional type of jackscrew was selected as the best choice because of the following advantages: it has a high load capacity for its size and weight, it is simple and reliable, it is self-storing in the blade, and its efficiency can be adjusted to a desired value by judicious selection of thread pitch and lubricant. A very high efficiency is neither needed nor particularly desirable because the rotor would tend to speed up when the blades are retracted due to the Coriolis effect. While operating at an overall mechanical efficiency of 50

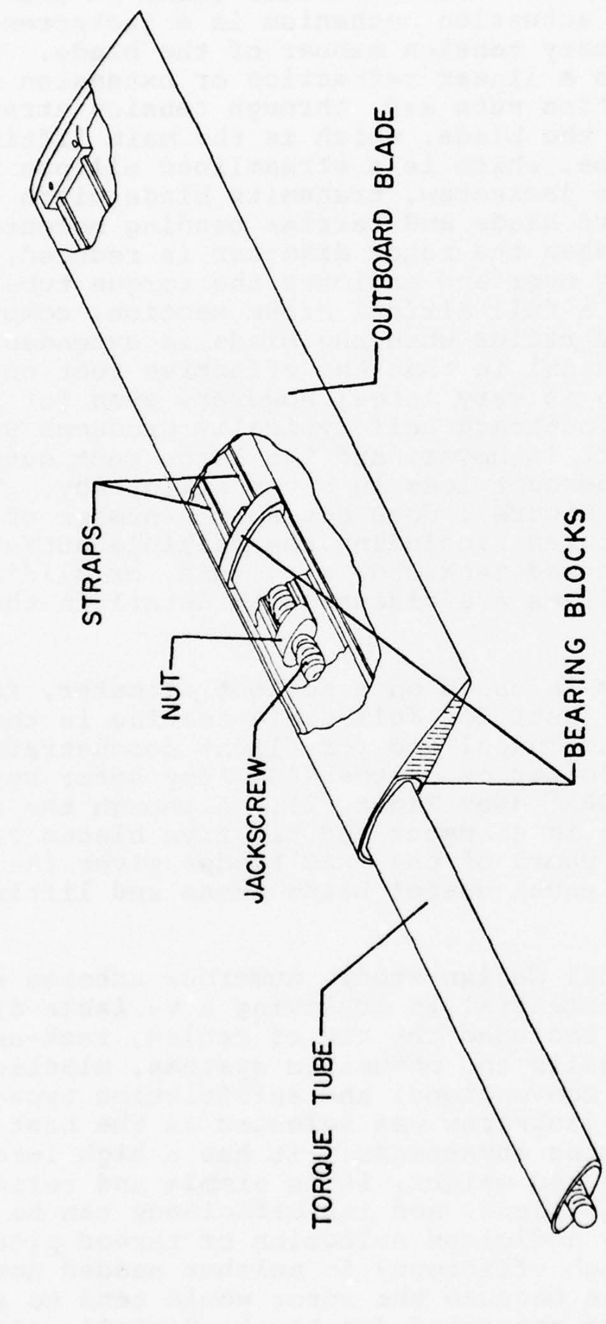

Figure 1. TRAC Rotor Blade Schematic.

Sikorsky S-61 Helicopter

Sikorsky RSRA Helicopter

Figure 2. Potential TRAC Rotor Applications.

percent, the rotational kinetic energy of the blades is dissipated at just the rate required for retraction at constant rpm in the absence of any external aerodynamic torque. A self-locking system results if the static efficiency is less than 50 percent. The jackscrew serves as a heat sink during diameter changes and can be cooled conveniently, if required, with centrifugally pumped air. The jackscrew, in combination with a rotor-head differential drive system, provides positive synchronization between blades, with no requirement for rachets, pawls, or gear-shifting devices. The jackscrew can be made structurally redundant (for high reliability) by adding an internal tension strap which will retain the blade in the event of a jackscrew fracture. In addition to being self-storing, the mass of the jackscrew contributes substantially to blade flapping stability at minimum diameter, allowing operation at a very high advance ratio.

The blade planform arrangement, wherein the outer blade slides over and encloses the torque tube, was selected over an alternate arrangement in which the outer blade slides into an airfoil-shaped sleeve. The sleeve configuration results in more blade surface area being exposed after retraction than the adopted configuration and has a severe structural problem related to carrying the blade bending moments, particularly chordwise moments, across the sliding joint. The selected configuration greatly diminishes the problems of the sliding joint and, despite the large root cutout, has a hovering efficiency nearly as high as a blade with the conventional planform.

DETAILED DESCRIPTION

Blade Assembly

A full-scale preliminary design study of a flightworthy TRAC rotor system, conducted under the sponsorship of the Eustis Directorate of the U. S. Army Air Mobility R&D Laboratory, is reported in Reference 5. An assembly drawing of the TRAC blade from Reference 5 is shown in Figure 3. Although the hardware fabricated under the present study is not identical to this design, it is similar in nearly all aspects important to this investigation. For example, the internal strap in the jackscrew, which provides structural redundancy in the

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[5] Fradenburgh, E. A., Hager, L. N., Kefford, N. F., EVALUATION OF THE TRAC VARIABLE DIAMETER ROTOR: PRELIMINARY DESIGN OF A FULL-SCALE ROTOR AND PARAMETRIC MISSION COMPARISONS, Sikorsky Aircraft Division, United Technologies Corporation; USAAMRDL Technical Report 75-54, Eustis Directorate, U. S. Army Air Mobility R&D Laboratory, Fort Eustis, Virginia, February 1976, AD023448.

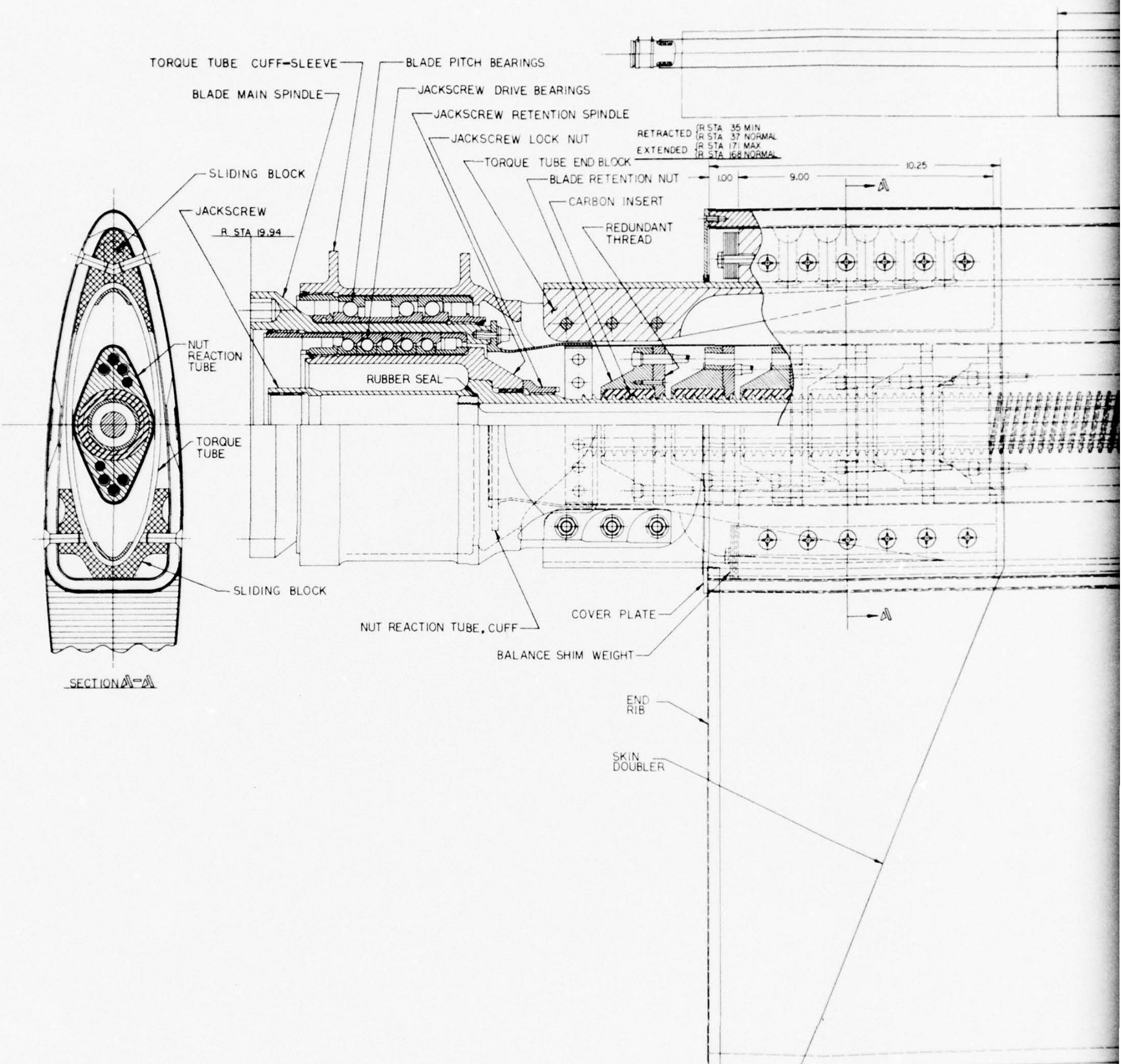
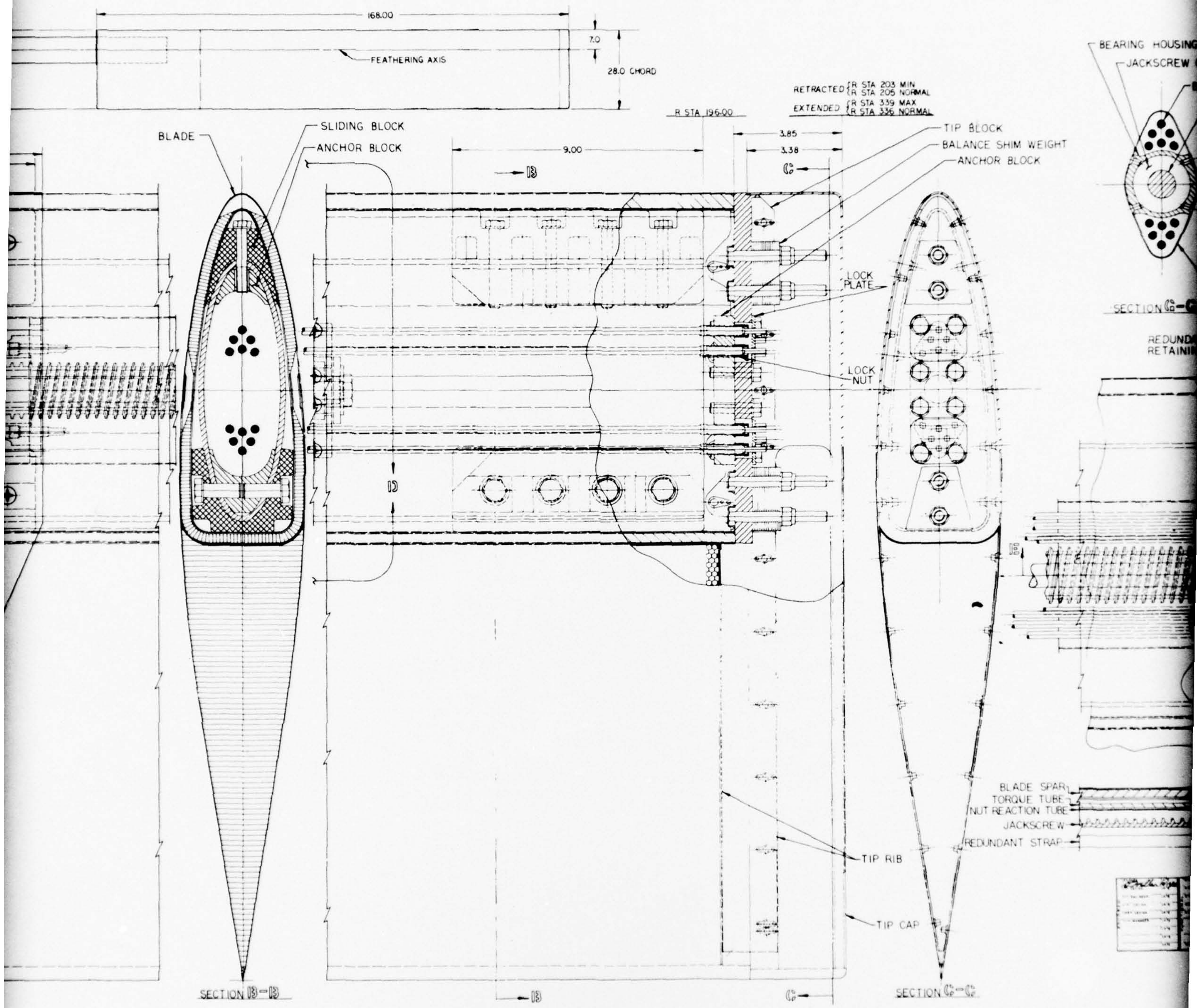


Figure 3. Flightworthy TRAC Rotor Blade Assembly.



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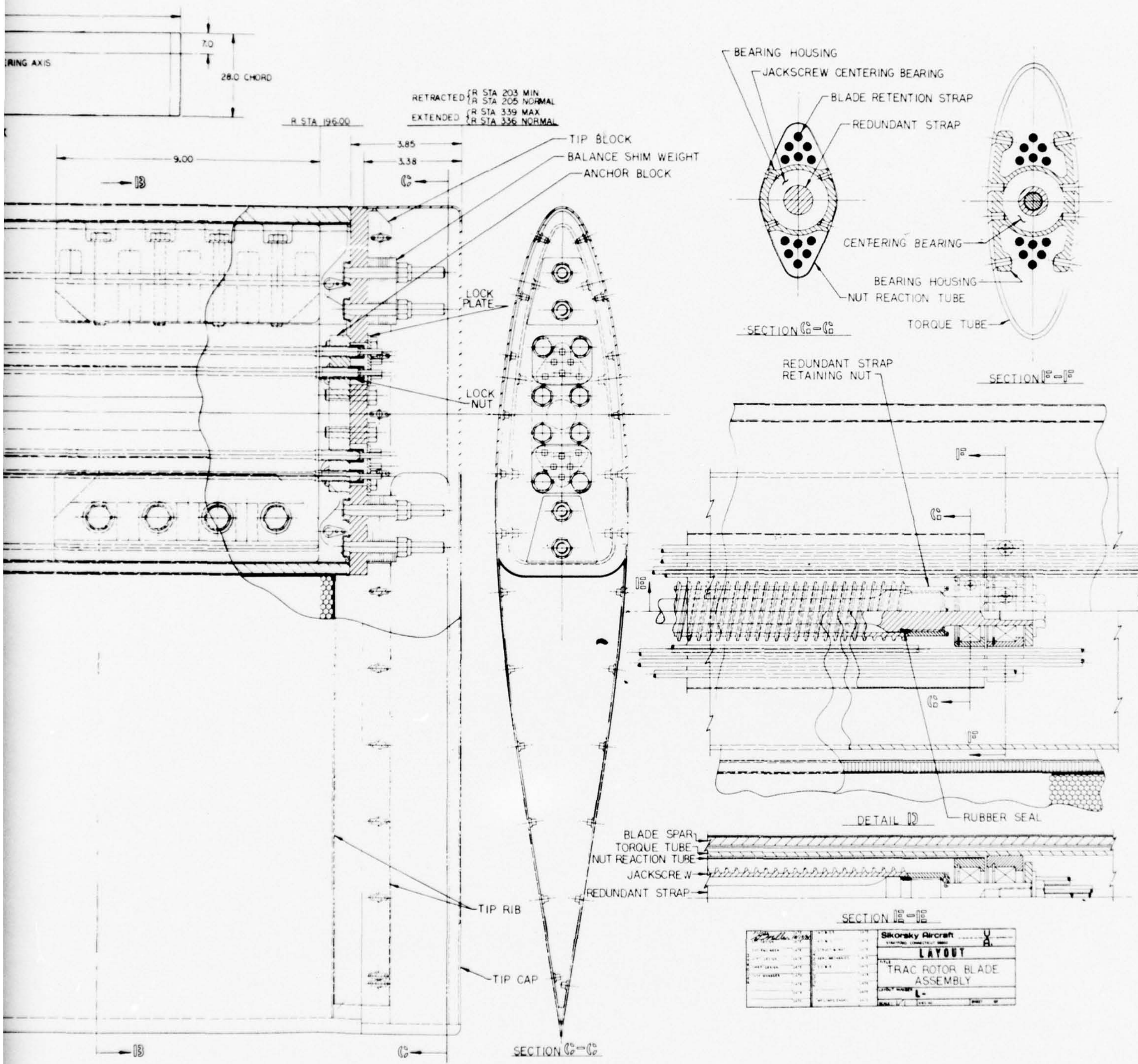

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SECTION G-G
SECTION F-F
REDUNDANT STRAP RETAINING NUT
RUBBER SEAL
DETAIL D
BLADE SPAR
TORQUE TUBE
NUT REACTION TUBE
JACKSCREW
REDUNDANT STRAP
SECTION E-E
Sikorsky Aircraft
LAYOUT
TRAC ROTOR BLADE ASSEMBLY

3

flightworthy design, was not included in the laboratory test articles because this would have no influence on the friction or wear properties of the screw. Similarly, the mounting and drive hardware for the jackscrew are different than shown in Figure 3.

The relationship of the blade components is illustrated in Figure 4. Some of the full-scale TRAC components, fabricated for the present investigation, are shown in Figure 5. These components are sized to represent the full-scale preliminary design shown in Figure 3.

Blade Spar

The design constraints on the TRAC outer blade spar are different from those of a conventional blade. One requirement is that the blade spar provide adequate space for the inboard blade components when the blade is telescoped. Another difference is that the TRAC spar is in compression rather than tension. This fact dictates that the blade must be stiff enough to avoid column buckling from the compressive forces and local buckling from the transverse bending loads. Thus, the TRAC outer blade spar utilizes a somewhat thicker than normal blade airfoil section and is designed for somewhat higher flapwise stiffness. The external dimensions of the blade spar are based on a 28-inch chord, 63_2A016 symmetrical airfoil section. Although the flightworthy design calls for either an aluminum honeycomb sandwich or a filament-wound graphite fiber composite construction, a simpler design was utilized in the present investigation: a single-piece closed D-section aluminum extrusion. This construction, although heavier than the Reference 5 design, was less costly to fabricate and has similar stiffness properties.

Torque Tube

The torque tube is an oval aluminum extrusion sized to fit inside the outer blade spar. This component transmits flapwise and chordwise bending loads and shear forces across the sliding joint to the blade root end, as well as transmitting blade pitch change motions from the push rods to the outer blade. The nut reaction tube described in Reference 5, which would fit inside the torque tube and react the nut friction torque generated during retraction and extension, was not represented in the present investigation. In the test program, a single, multi-purpose torque tube was to be used. This torque tube and the blade spar are shown being fitted in Figure 6.

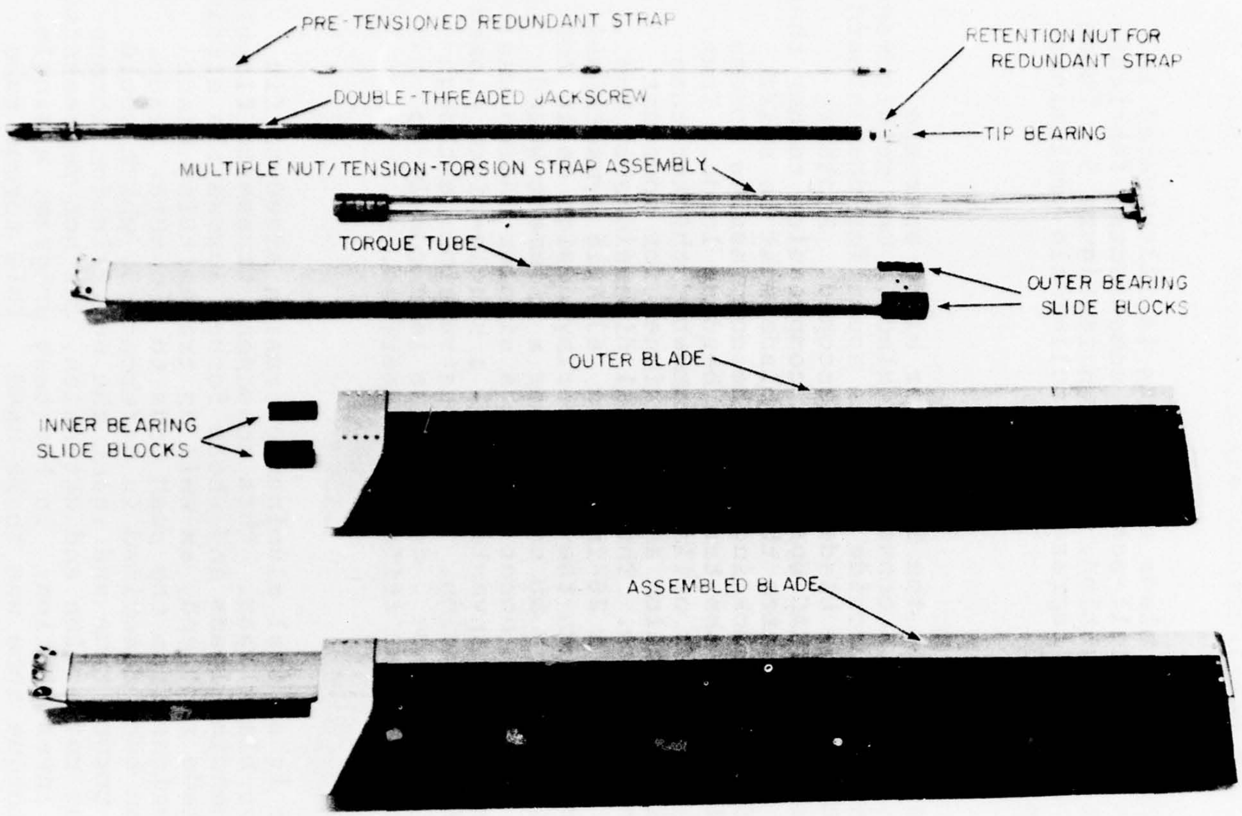


Figure 4. TRAC Dynamic Model Rotor Blade Components.

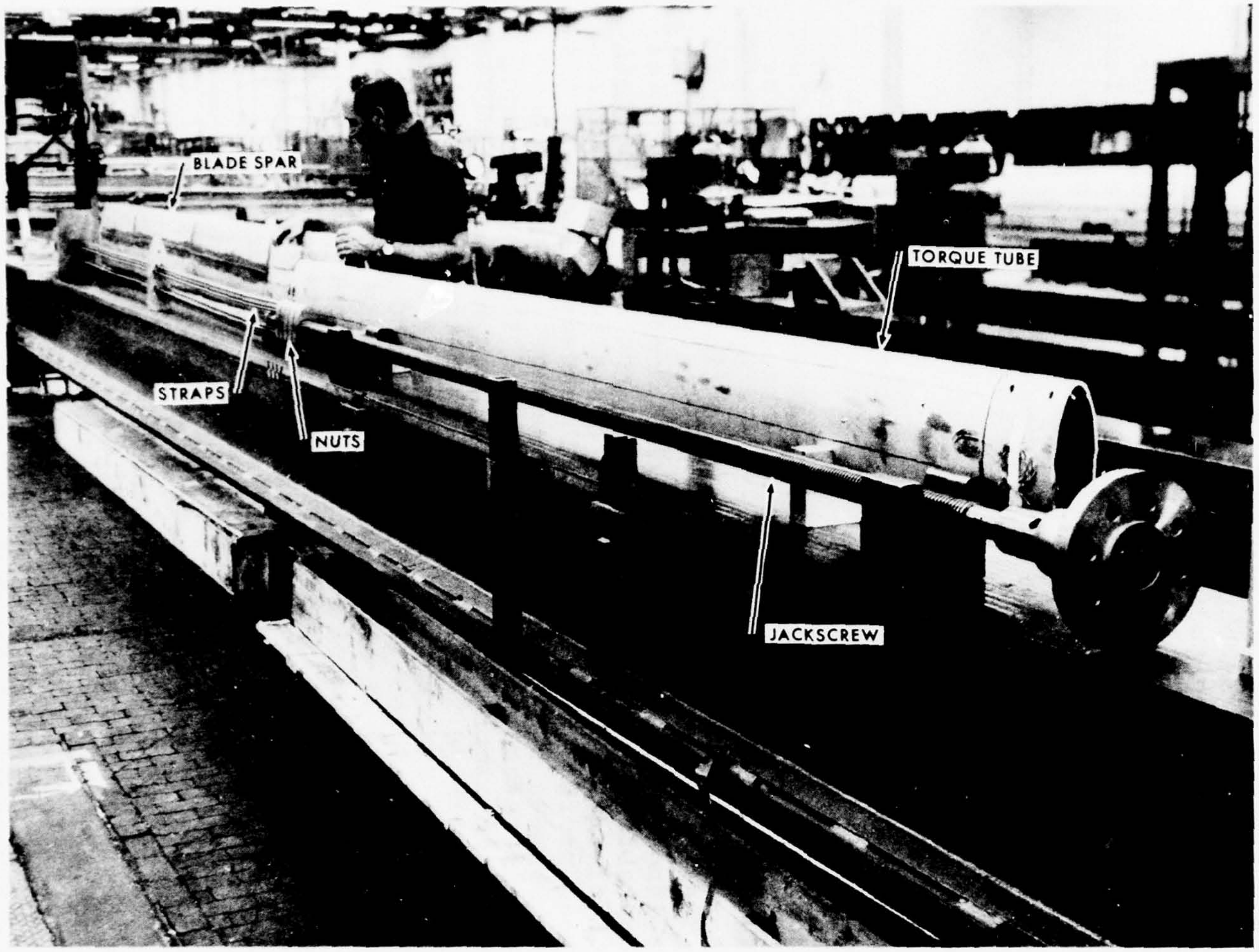


Figure 5. Full-Scale TRAC Test Components.

Bearing Blocks

The bearing blocks are attached to the outboard end of the torque tube and to the inboard end of the blade spar and are shaped to bear against the inner and outer contours, respectively, of the mating members (see Figures 3 and 6). These blocks provide the longitudinal bearing surfaces for the telescoping components. In the present investigation, the blocks were fabricated from aluminum, with either Teflon or nylon pads bonded to the bearing blocks to provide a lubricated surface and minimize wear. This method was chosen over a block of solid bearing material because it provides the possibility of shimming to adjust for manufacturing tolerances and wear and because it can be easily replaced.

Jackscrew

The jackscrew has a double thread (sometimes referred to as a "two-start" thread) because the resulting increased thread pitch angle increases screw efficiency. The extended thread lead also reduces the time for blade retracting or extension. The jackscrew has a left-hand thread because this makes the outer shaft of a coaxial diameter-change shaft system (inside the main rotor shaft in the aircraft configuration) serve the function of the retraction shaft. Because blade retraction requires larger mechanical torques than does blade extension, it is desirable to make the outer shaft the retraction shaft. In the final test configuration, the jackscrew has a buttress-type thread profile with the thread thickness being less than the space between the threads. This design minimizes the volume and therefore the weight penalty of the threads and also permits considerably greater thickness on the nut threads, which are fabricated from lower-strength materials and which are subjected to greater wear rates than the screw materials. The outboard end of the jackscrew is centered relative to the torque tube by means of a needle bearing whose housing is bolted to the torque tube. This arrangement permits the required degree of relative axial motion between the torque tube and the jackscrew while maintaining the correct transverse position. This bearing housing is shown in Figure 7.

Nuts

In the final test configuration, there are six nuts that transfer the centrifugal force from 12 blade-retention straps to the jackscrew. Nut number 1, closest to the rotor head, has one pair of holes, counterbored to retain its pair of straps. These straps pass freely through holes provided in the other five nuts. Nut number 2 has two pairs of holes, one pair of which is to retain its own pair of straps and one pair of which is to allow straps from nut 1 to pass through. The same principle applies to all nuts, and nut number 6, closest

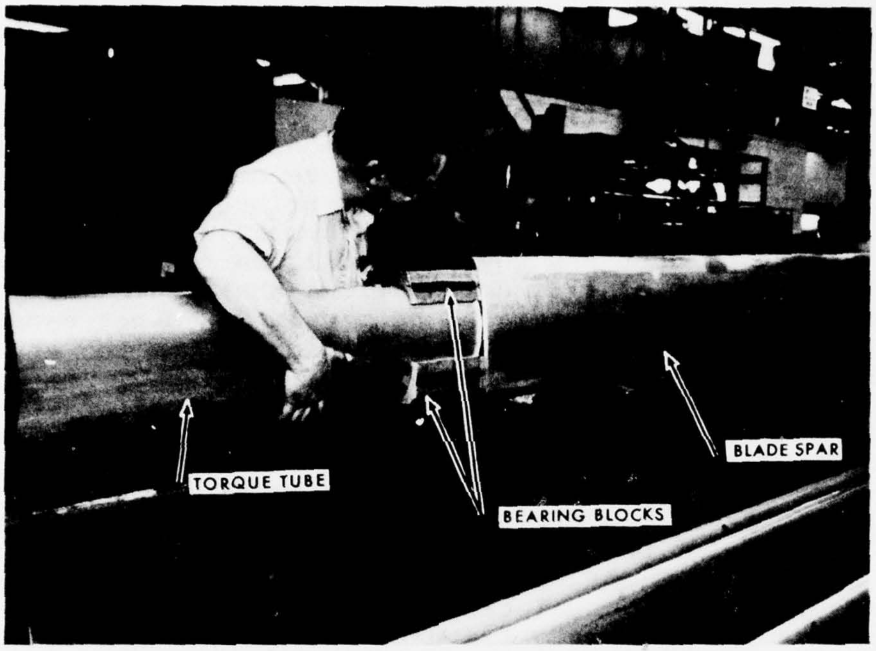


Figure 6. Full-Scale Torque Tube and Outer Blade Spar.

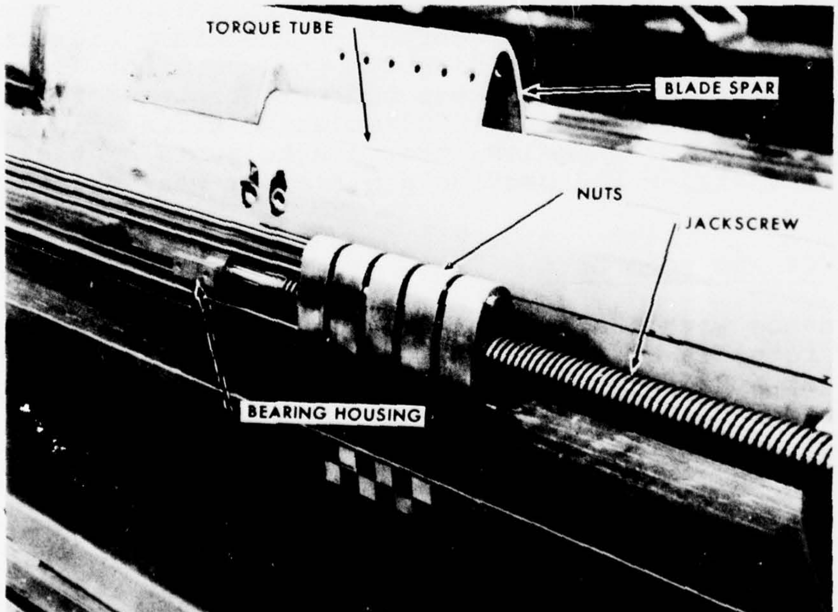


Figure 7. Initial Jackscrew/Five-Nut/Strap Configuration.

to the blade tip, has six pairs of holes. The two retention straps for each nut are always diametrically opposite each other to avoid any moment transfer from the straps to the nuts.

The reason for the multiple nut/strap configuration, which is similar to that used on the dynamic model blade, is to provide structural redundancy in case of failure of any component and also to improve the thread loading distribution. If a single nut were to be provided with a large number of threads, only a few threads on each end would actually carry a significant loading because of elastic deformations that occur in both screw and nut. The threads in the middle would not carry much load. This would lead to the overloading of the few end threads and excessive wear or progressive static failure. The problem is minimized by dividing the nut into a number of segments with fewer threads each. The strap geometry ensures that each nut carries a full share of the load because the elastic stretch of the strap under load is large compared to initial errors in providing precise strap lengths, particularly when the straps are "tuned" on the bench; i.e., where the assembly is loaded to part load and the strap lengths are adjusted as necessary to provide equal tension. The blade assembly incorporates the necessary features to provide these adjustments. The twelve tension straps are threaded at the tip and anchored into a drilled and tapped tip block.

The jackscrew/nut assembly features a "centralizing" acme thread design. In this configuration, the minor diameter of the screw serves as a pilot diameter for the minor diameter of the nut, thus tending to keep the nuts centered relative to the screw during loading. Sufficient clearance is maintained at the major diameter interface to avoid contact in the nut root fillet and provide a place for wear particles to accumulate.

PERFORMANCE AND LOAD DATA

In accordance with the preliminary design study of Reference 5, the TRAC rotor is designed for a steady-state lift capability in hover at 4000 feet, 95°F, of approximately 20,000 pounds. The criterion selected for the analysis was a rotor lift of 21,000 pounds to provide a net lifting capability of 20,000 pounds with an airframe vertical drag allowance of 5 percent.

A diameter of 56 feet (extended) was selected as appropriate for the design requirements. The rotor diameter in the retracted position will be 33.6 feet (40-percent reduction). An outer blade chord of 28 in. was selected to provide a blade aspect ratio identical to those of the preliminary design and the dynamic model reported in USAAMRDL Technical Report 73-32, Reference 1, because of the decision to not deviate substantially from this successful baseline design. These dimensions provide a useful blade area ($b\bar{c}R$, where b is the number of blades, $\bar{c}$ is the thrust-weighted mean chord and R is the extended radius) approximately the same as that of the standard H-3 rotor system, which has a useful lifting capacity of 20,000 pounds or somewhat more, depending on the mission.

A comparison between the TRAC rotor and other Sikorsky rotor systems in the same general size category is shown in Table 1, taken from Reference 5. It can be seen that in most respects the TRAC rotor has dimensional and blade loading parameters which are typical of other helicopter systems. The important parameter of dimensionless blade loading, C_T/σ, is essentially the same as for the H-3 and H-34 aircraft. The Sikorsky YUH-60A UTTAS is a higher performance pure helicopter than the other two and for this reason uses lower values of C_T/σ. Because the TRAC rotor will be used in aircraft configurations having wings and auxiliary propulsion systems, it is not necessary to utilize low C_T/σ values to achieve high speeds. Thus, the present design is conservatively sized for a 20,000-pound lift capability. Rotor thrust at the normal design tip speed at 4000 feet, 95°F, for a C_T/σ value of 0.12, is over 24,000 pounds.

A tip speed of 660 ft/sec (225 rpm) was selected for the design. This tip speed not only provides adequate hovering performance, but also flight as a pure helicopter (without benefit of wing lift or auxiliary propulsion) up to a forward speed of 100 knots.

The design of the blade was an iterative process in which various assumptions of loads and blade properties were made and then modified as required to provide adequate strength and minimum weight. The description of the blade's internal components is limited to the final configurations and ignores the iterative process.

In the aircraft configuration, the centrifugal load distribution in the various blade components at full blade extension is shown in Figure 8. The outboard portion of the blade starts at the 50% radial station (14 feet) and extends to the tip. This component is in compression. At the tip, this centrifugal load is picked up by the retention straps and is delivered through the set of six nuts to the jackscrew and finally to the sleeve/spindle assembly, with each of these components adding its own centrifugal load. The torque tube is independently supported by the sleeve/spindle assembly. The centrifugal load distribution with the blade components in the retracted position is shown in Figure 9.

The final TRAC design data is summarized in Table 2.

TABLE 1. COMPARISON OF TRAC ROTOR AERODYNAMIC PARAMETERS WITH OTHER ROTOR SYSTEMS

	TRAC ROTOR PRELIM. DESIGN	H-3 (S-61)	H-34 (S-58)	YUH-60A (UTTAS)
Diameter (2R), ft	56 (fully extended)	62	56	53
Disk area (πR^2), ft^2	2463	3019	2463	2206
Number of blades (b)	4	5	4	4
Blade twist, θ_1, deg	-8 (fully extended)	-8	-8	-16
Mean blade chord ($\bar{c}$), ft	2.14	1.52	1.37	1.73
Blade aspect ratio ($\frac{R}{\bar{c}}$)	13.1	20.4	20.5	15.3
Solidity $\frac{b\bar{c}}{\pi R}$	.0972	.0780	.0621	.0831
Effective blade area ($b\bar{c}R$), ft^2	239.4	235.7	153.1	183.3
Typical Rotor Lift (L), lb	21,000	21,000	13,000	16,000
Disk Loading ($\frac{L}{\pi R^2}$) lb/ft^2	8.5	7.0	5.3	7.3
Blade Loading ($\frac{L}{b\bar{c}R}$), lb/ft^2	87.7	89.1	84.9	87.3
Normal Tip Speed (ΩR), ft/sec	660	660	649	730
$C_T/\sigma = L/\pi R^2 \rho (\Omega R)^2 \sigma$				
(a) Sea Level Standard	.0847	.0861	.0848	.0689
(b) 4000 ft 95°F	.1048	.1065	.1049	.0852

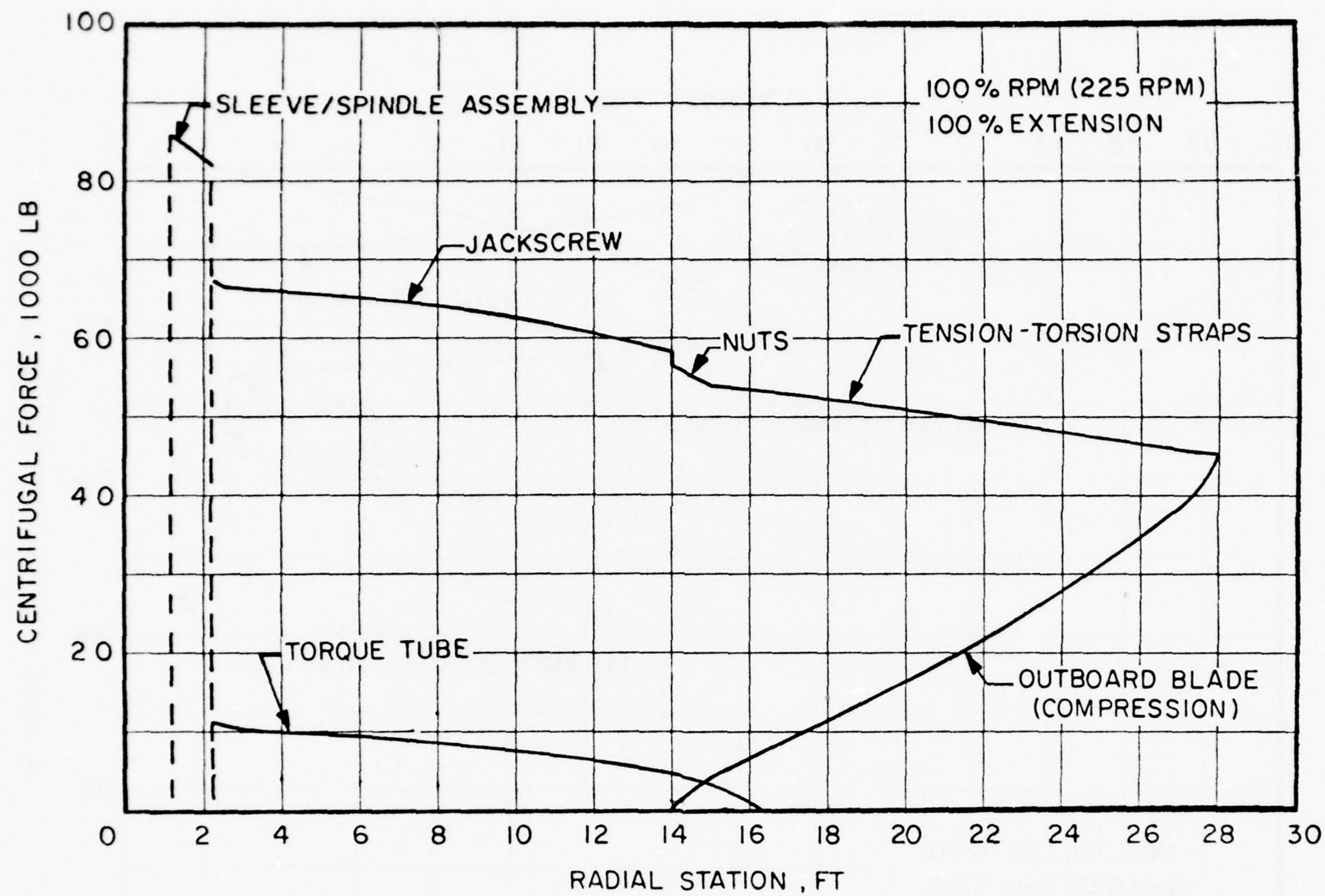


Figure 8. Blade Centrifugal Loads - Extended Position.

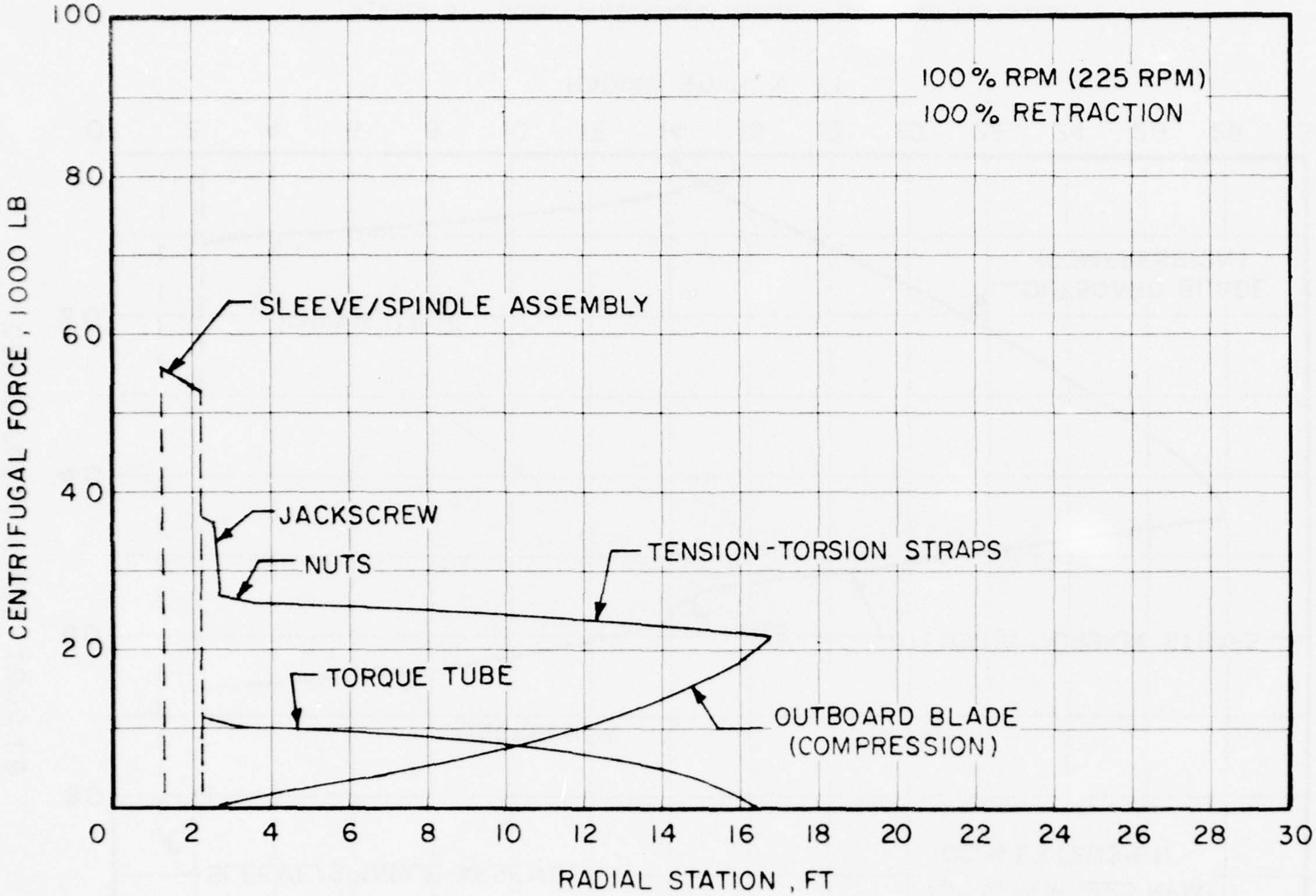


Figure 9. Blade Centrifugal Loads - Retracted Position.

TABLE 2. TRAC DESIGN DATA

Rotor Design Data

Rotor Dia = 56 ft (extended); 33.6 ft (retracted)
Outer Blade Length - 14 ft
Outer Blade Chord = 2.14 ft
Number of Blades = 4
RPM = 225
Tip Speed = 660 fps
Hover Out of Ground Effect - 4000 ft 95^{o}F @ 20,000 lb gross wgt
Blade Retraction Time = 36 sec
Nominal Aircraft Speed @ Retraction = 150 knots
Rate of Retraction = 3.75 in/sec
Blade Lift at Retraction = 0

Jackscrew and Nut

Dynamic Efficiency = 50%
Thread - Left Hand (double thread)
RPM (Jackscrew) = 337.5
Nut Travel = 134.2 in
Centrifugal Loads at Nuts
 Retracted = 26,300 lb
 Extended = 56,600 lb

Retention Straps

Centrifugal Load/Strap = 4720 lb (max)

Torque Tube

- Centrifugal Force = 64,900 lb (retains blade in event straps fail)
- Elliptical Tube Shape: major diameter = 9.33 in; minor diameter = 3.11 in

Blade Spar

- Centrifugal Force = 45,100 lb
- Airfoil 63_2016 (modified)

LABORATORY MATERIALS RESEARCH PROGRAM

Early in the program, two full-scale jackscrews and two mating nut assemblies were fabricated from the following materials:

Jackscrew	Nut
Custom 455 stainless steel coated with Vitrolube NP1-1220 dry-film lubricant	A286 stainless steel nitrided and coated with Union Carbide LCN-1
Custom 455 stainless steel coated with Dicronite DL-5	A286 stainless steel nitrided and coated with Vitrolube NP1-1220

This arrangement provided four possible jackscrew-nut combinations. These are the designs shown in Figures 5 and 7.

During the initial break-in test runs of the first jackscrew and nut combinations (jackscrew coated with Dicronite DL-5, nut coated with LCN-1), severe scoring and galling of the jackscrew occurred at relatively light loads accompanied by intermittent squealing. The second set of nuts was then installed at an unworn section of the screw and the straps loaded to approximately 7000 lb total. The screw was rotated at low speed (20 rpm) for approximately one minute. Excessive wear and pickup of the screw material occurred again at the outside corner of the screw thread (see Figure 10).

As a result of these initial tests, it became apparent that further development of the jackscrew-nut configuration was necessary before comparative testing could continue. Attempts at lapping the nuts and jackscrew together proved unsuccessful. Metal pickup and scoring of the jackscrew continued to be the problem. Obviously, line contact, instead of area contact, was taking place and possibly the nut and screw materials were not compatible for the desired operating conditions. It was concluded, at this point, that it was necessary to conduct a basic laboratory materials research program to determine wear and temperature characteristics of a variety of screw and nut material combinations preparatory to choosing the best material for the final jackscrew and nut design.

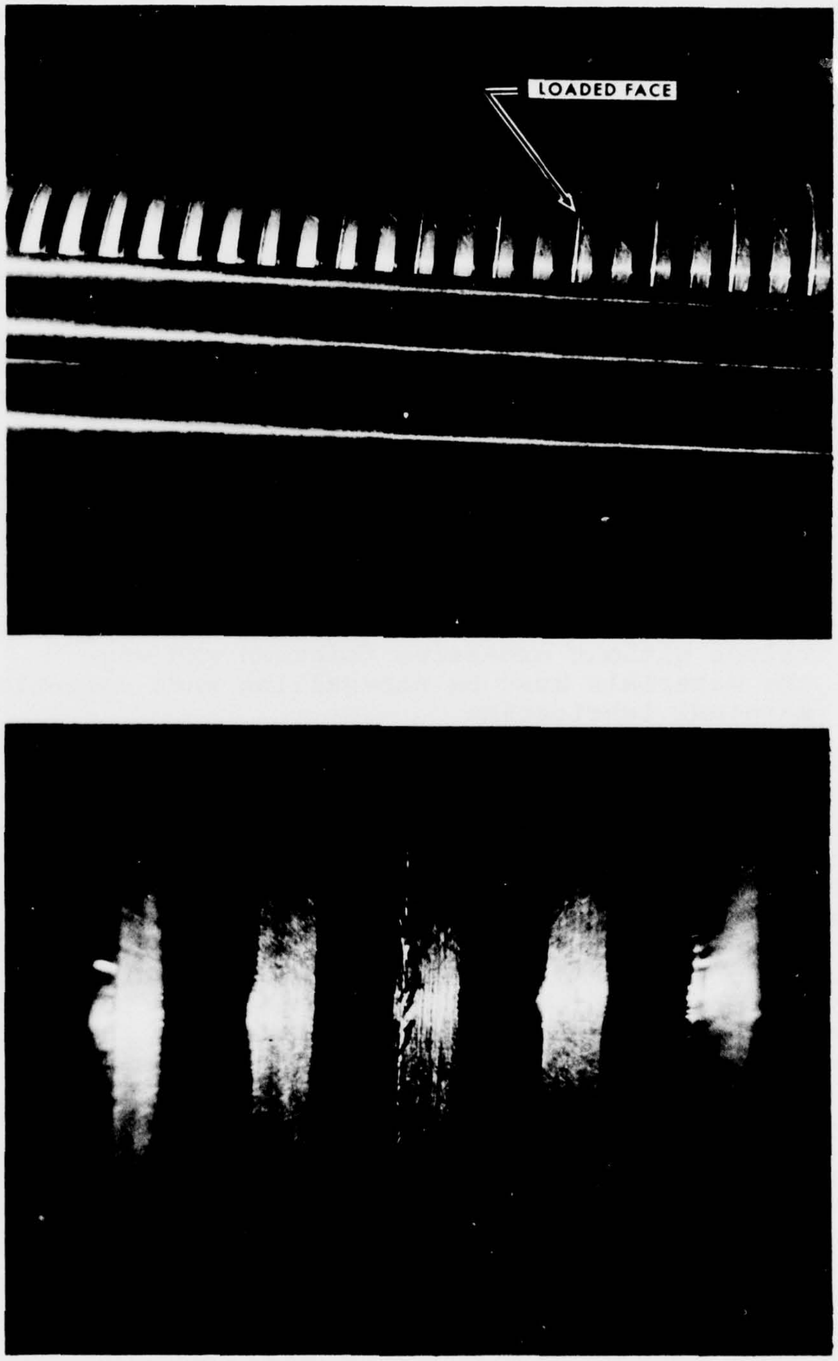


Figure 10. Jackscrew Condition after Initial Testing.

The experience with the initial jackscrew/nut design showed that a crucial element in the successful operation of the TRAC extension/retraction system is the selection of a suitable material combination for the jackscrew and the mating nuts. Since these components must be small enough to fit within the torque tube envelope, they will be subjected to very high "PV" values. PV is the product of the normal pressure on the thread surfaces in psi and the rubbing speed in fpm. The TRAC max PV value cannot be estimated precisely, because the actual contact pressures depend on the elastic and thermal deformations in the jackscrew and nut threads, but probably is on the order of one million for the most highly loaded threads. This value is within the upper fringe of current jackscrew-nut technology, and thus, a critical revaluation of candidate rubbing materials was considered necessary. Design requirements for the rubbing pair were that:

- the system must be self-lubricating (light grease or oil would be thrown outboard in a whirling blade)
- the material pair must be compatible with the aircraft operating environment
- the materials must be able to operate at the high PV values without excessive friction and wear
- the materials must be non-galling when operating with marginal lubrication
- the materials must be capable of precision fabrication into fairly complex shapes.

To satisfy the above requirements, a materials bench test program was initiated in concert with the MAIC Division of the Pure Carbon Co, Saint Marys, Pennsylvania, to study various self-lubricating rubbing material pairs in a simulated jack-screw and nut configuration under conditions of high PV values. The primary objective of this program was to screen a significant number of promising material combinations to select two pairs which had the best chances of satisfying the TRAC design requirements.

These tests utilized rings that simulated a single thread contact. It was realized at the outset that the ring tests could not duplicate exactly the jackscrew-nut situation, but it was felt that the results would be qualitatively representative. If anything, the ring tests are more severe because the generated heat has little place to go, whereas, in the actual situation, the nut is continuously advancing onto cold steel, which acts as a heat sink.

Test Specimens

The nut ring-test specimens and the screw ring-test specimens were based on a two-inch-diameter screw thread and were designed so that the actual TRAC operating parameters (pressure and rubbing speed) would be simulated in the test as closely as possible. These specimen designs are shown in Figure 11.

Candidate Materials

After a materials search of existing jackscrew and nut designs and after consultation with recognized experts in the field of friction and wear, the choice of candidate materials and lubricants was narrowed down to those shown in Table 3. The combination of materials that had proven to be unsatisfactory in the initial screw/nut tests was included in the ring test program as a reference.

TABLE 3. CANDIDATE MATERIALS.

Specimen Part Number	Material
	Screw
8A	Carpenter Custom 455 stainless steel
8B	AMS-5643 (17-4PH) stainless steel with electrolizing chrome plate.
8C	VASCO 300 CEVM maraging steel
8D	MIL-S-6709 (Nitralloy 135-M)
8E	VASCO 300 CEVM with electrolizing chrome plate.
	Nut
10A	P-3310 carbon graphite (Pure Carbon Company)
10B	P-59-L leaded bronze (Pure Carbon Company)
10C	PM-103 Molalloy (Pure Carbon Company)
10D	Berylco-25 beryllium copper
10E	leaded bronze (cast)
10F	Mueller brass alloy 721-E
10G	AMS-5737 (A-286) stainless steel (nitrided)
13C	PM-103 heat shrunk into metal casing.

TABLE 3. (Continued)
Lubricants
Vitrolube dry film - National Process Industries
Electrofilm 5306 (MIL-L-8937-A) - Electrofilm Inc
Tribolube 1 (MIL-G-83363) - Aerospace Lubricants (grease)
Tribolube 2 (MIL-G-83261) - Aerospace Lubricants (grease)

It can be appreciated at this point that not all possible combinations of the materials in Table 3 could be tested; the matrix of 38 combinations actually tested in the initial screening tests is identified in Figure 12.

Test Rig Description and Test Procedure

The test rig was designed and fabricated by the MAIC Division of the Pure Carbon Company, Inc. The test rig is mounted on a 30-horsepower, variable-speed-drive test stand. Test speeds ranged from 200 to 600 rpm. Interposed between the test rig and the variable-speed drive is an electromagnetic clutch, which is used to achieve the rapid engage and disengage cycle required by the application. The test rig installation is shown in Figure 13. In this installation, the screw test specimen is the rotating member whereas the nut test specimen is restrained against rotation by a 5-inch-long torque arm. The entire rig drive system is interlocked into the electromagnetic clutch and controlled by a set of cycle and delay timers and a temperature controller. The thrust load is applied to the test specimens by a hydraulic cylinder acting through a precision ball bearing. The ball bearing provides a low-friction mounting for the nut test specimen, allowing reaction torque to be conveniently measured. The test rig is capable of applying thrust loads up to 10,000 pounds. During the initial test runs, a direct reading force gage was used to measure the reaction torque. This was later replaced by a load cell that continuously recorded reaction torque on one channel of an oscillograph recorder. The test stand was also instrumented to record data on nut specimen temperature and screw specimen temperature.

The tests were run with the unit loading applied continuously. The rotation was initiated by clutch engagement and held for the "on" cycle time, followed by clutch disengagement. A cooling period was then allowed while the heat generated by friction was dissipated into the surroundings. When the measured temperature of the specimens dropped to 150^{o}F, the temperature controllers automatically reengaged the clutch for the next "on" cycle. The cooling period, which varied from a few seconds to a minute or more - depending on the particular test conditions - , was not counted as part of the total test time.

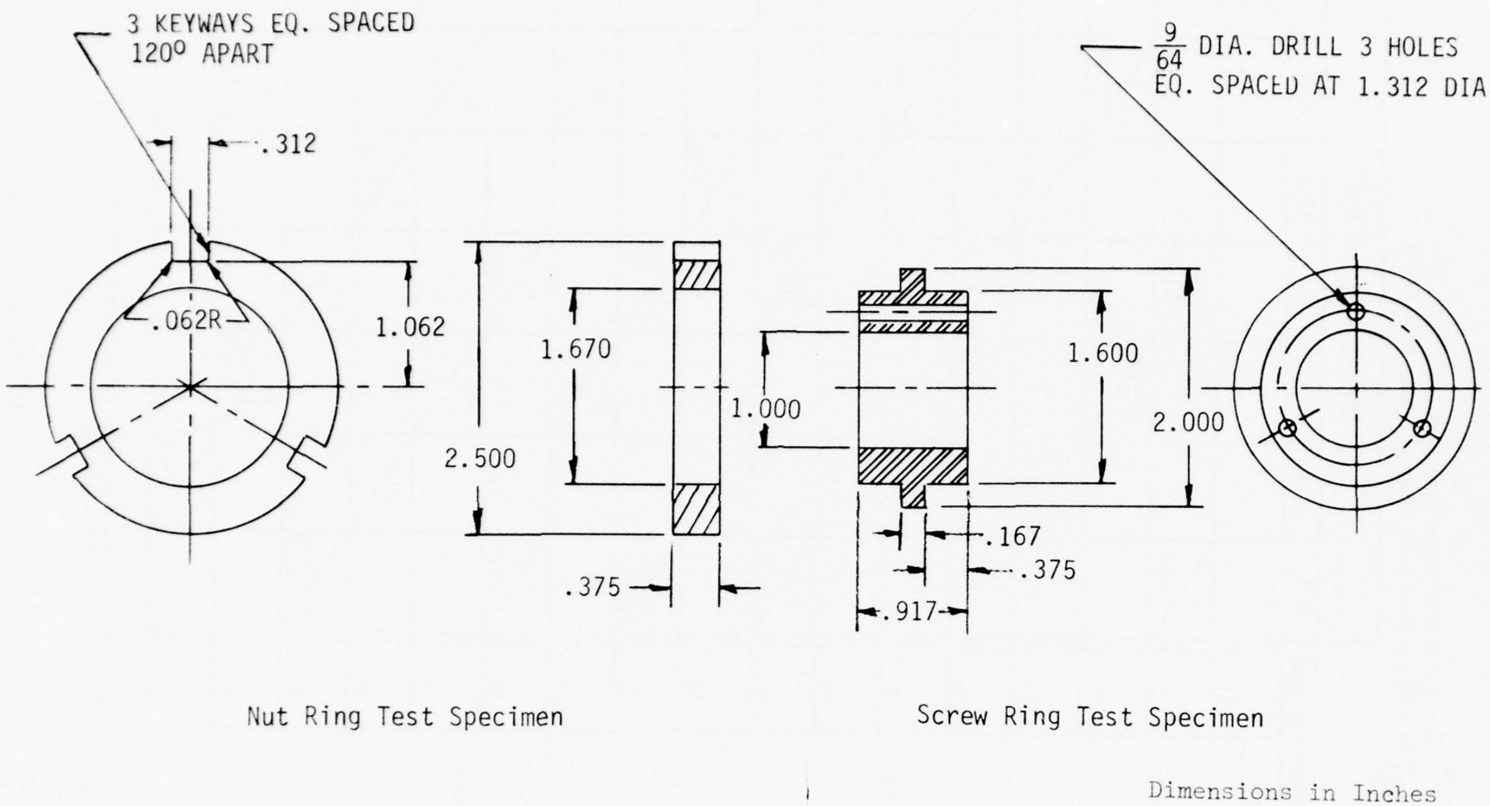


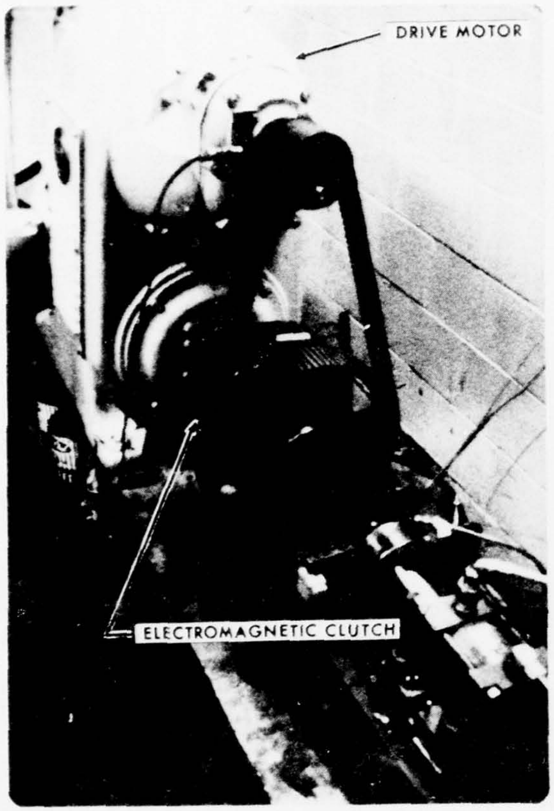
Figure 11. Ring Test Specimens.

Code: Test No.* / X-XX   X = Final Test Point Completed (See Table 4)
XX = Total Time in Seconds at Final Test Point

SCREW SPECIMENS FOR TEST**		NUT SPECIMENS FOR TEST**						
LUBE USED		10 A	10 B	10 C	10 D	10 E	10 F	10 G
VITROLUBE	8A							31 / 3-10
	8B	22 / 3-600			20 / 3-600			
	8C			24 / 3-15			4 / 2-205; 5 / 3-300	
	8D		23 / 3-600			21 / 3-50		
ELECTROFILM	8A							30 / 2-175
	8B			25 / 2-300			28 / 3-10	
	8C		19 / 3-47			6 / 2-250		
	8D	26 / 3-405			38 / 3-197; 27 / 3-270			
MIL-G-83363	8B		18 / 2-300			8 / 3-42		
	8C	17 / 2-165			2 / 2-20; 7 / 2-35			
	8D			32 / 2-45			29 / 2-9	
MIL-G-83281	8B	16 / 1-120			9 / 2-14			
	8C			15 / 2-300			10 / 2-8	
	8D		33 / 3-105			34 / 3-35		
NONE	8B	36 / 2-40; 14 / 1-120	37 / 2-30		11 / 2-8			
	8C			3 / 2-21; 13 / 2-130			12 / 2-5	
	8D		35 / 3-195					

*See Table 5
**The materials are identified here by part number. Refer to Table 3 for material identification.

Figure 12. Ring Test Matrix and Results.



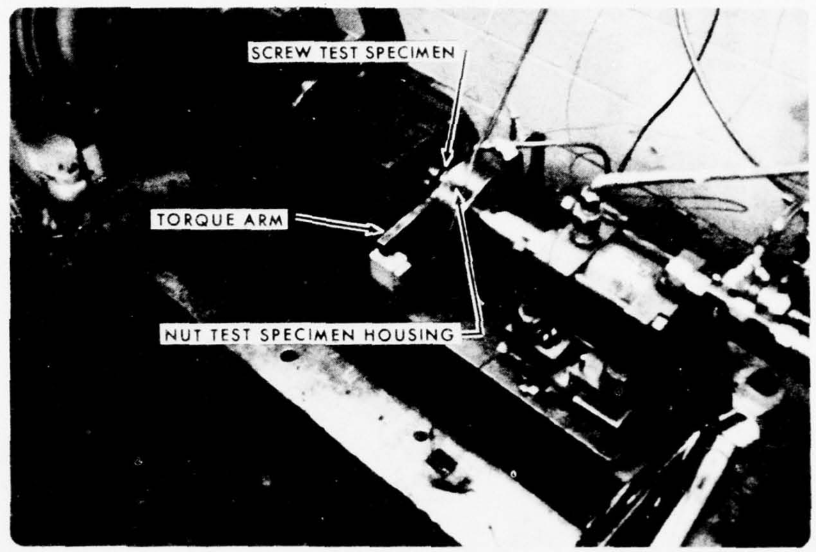


Figure 13. MAIC Ring Test Rig.

Test Program

The test program was conducted in two phases. Phase I was a screening test conducted to preliminarily evaluate the various combinations of test specimen materials and lubricants listed in Table 3. Phase II was designed to subject the most promising material combinations from Phase I to more stringent test conditions in order to obtain additional data and achieve final ranking. A thermal gradient test and an overload test were also included as part of Phase II.

Phase I Testing

For the initial screening tests, the test specimens were subjected to three load conditions, identified in Table 4 as test points 1, 2, and 3. If the material combination being tested survived test point 1, it was then tested at the load conditions of test point 2, and then, if again successful, to the load conditions of test point 3. The material combinations for these initial tests are shown in Table 5 (test numbers 1 through 4).

TABLE 4. TEST CONDITIONS FOR INITIAL SCREENING TESTS

Test Point	Unit Load (psi)	Rubbing Velocity (fpm)	PV	"On" Cycle Time (sec)	Total Test Time (min)
1	1000	100	100,000	5	10
2	3000	145	435,000	5	10
3	6000	190	1.14×10^6	5	10

Note: In tests involving carbon graphite or "Molalloy", only the metal screw specimen was processed with a dry film lubricant. ("Vitrolube" or "Electrofilm"). In the tests where both parts were metallic, the dry film lubricant was applied to both the nut and the screw test specimens. In tests involving either grease, MIL-G-83261 or MIL-G-83363, instead of the dry-film lubricant, the grease was applied uniformly to the screw specimens rubbing surface before the start of testing at each test point.

After the first four test runs, it became obvious that these initial test conditions would not discriminate between "good" and "bad" materials on an optimum basis, e.g.; materials were failing at test point 1 that were known to be capable of performing at much higher "PV" conditions. As a result, the test conditions of Table 4 were revised to reduce the total test time at the lower "PV" values. The revised test conditions are shown in Table 6 (test numbers 5 through 38).

TABLE 5. SUMMARY OF SCREENING TEST DATA.

Test No.	Nut P/N	Screw P/N	Lubricant	Testing Completed* (pt-sec)	Wear (.001 in.)** Nut	Wear (.001 in.)** Screw	Torque*** (pt/in.-lb)	Remarks
1	10A	8C	None	1-600	N/M	N/M	1/500	Excessive wear.
2	10D	8C	MIL-G-83363	2-20	17	(-4)	1/375	Ran 600 sec at point 1. Seized at point 2. Sheared drive pins.
3	10C	8C	None	2-21	N/C	N/C	1/45	Ran 600 sec at point 1 - Good condition.
4	10F	8C	Vitrolube	2-205	4	5	1/35	Ran 600 sec at point 1.
5	10F	8C	Vitrolube	3-300	6	3	3/125	Ran 300 sec at points 1 and 22
6	10E	8C	Electrofilm	2-250	1	N/C	1/90	Ran per revised test plan.
7	10D	8C	MIL-G-83363	2-35	2	(-4)	1/150	Nut rough and groved.
8	10E	8B	MIL-G-83363	3-42	6	1	2/500	Numerous surface gouges in nut.
9	10D	8B	MIL-G-83261	2-14	8	(1)	1/350	
10	10F	8C	MIL-G-83261	2-8	4	N/C	1/175	
11	10D	8B	None	2-8		(-5)	1/750	Nut rough and grooved.
12	10F	8C	None	2-5	6	(-7)	1/500	Nut surface very rough.
13	10C	8C	None	2-130	N/C	N/C	1/25	Nut broke/2 pieces/ID chiped
14	10A	8B	None	1-120	66	N/C	1/350	Nut cracked through anti-rotation slot.
15	10C	8C	MIL-G-83261	2-300	2	3	2/575	Nut broke/2 pieces/ID chipped
16	10A	8B	MIL-G-83261	1-120	136	3	1/325	Nut cracked through anti-rotation slot.
17	10A	8C	MIL-G-83363	2-165	1	(-1)	1/300	Nut broken into many pieces - severe wear.
18	10B	8B	MIL-G-83363	2-300	14	(-3)	2/250	Nut cracked.
19	10B	8C	Electrofilm	3-47	2	2	2/150	
20	10D	8B	Vitrolube	3-600	4	7	3/375	Good condition.
21	10E	8D	Vitrolube	3-50	4	11	2/175	Wore through Vitrolube.
22	10A	8B	Vitrolube	3-600	3	5	3/300	Good condition.
23	10B	8D	Vitrolube	3-600	4	3	3/350	Good condition
24	10C	8C	Vitrolube	3-15	2	9	2/350	Large chip in nut at thermocouple hole.
25	10C	8B	Electrofilm	2-300	(1)	3	1/50	Nut broken into many pieces.

TABLE 5. (Continued)

Test No.	Nut P/N	Screw P/N	Lubricant	Testing Completed* (pt-sec)	Wear (.001 in.)** Nut	Screw	Torque*** (pt/in.-lb)	Remarks
26	10A	8D	Electrofilm	3-405	N/C	4	2/130	Wore through Electrofilm
27	10D	8D	Electrofilm	3-270	N/C	7	2/150	Wore through Electrofilm.
28	10F	8B	Electrofilm	3-10	4	7	2/115	Nut and screw welded together
29	10F	8D	MIL-G-83363	2-9	9	(-2)	1/500	Nut material transferring to screw.
30	10G	8A	Electrofilm	2-175	1	2	1/95	Large wear scars on nut.
31	10G	8A	Vitrolube	3-10	5	12	2/100	Wore through Vitrolube.
32	10C	8D	MIL-G-83363	2-45	3	2	1/25	Nut chamfered .030/.040 in. and no thermocouple.
33	10B	8D	MIL-G-83261	3-105	3	1	2/225	Nut cracked through the anti-rotation slot.
34	10E	8D	MIL-G-83261	3-35	4	N/C	2/625	
35	10B	8D	None	3-195	3	3	2/250	Nut cracked through the anti-rotation slot.
36	10A	8B	None	2-40	1	N/C	1/200	Special test - screw surface polished.
37	10B	8B	None	2-30	N/M	N/M	1/200	Special test - ran 70 sec at point 2 conditions.
38	10D	8D	Electrofilm	3-197	N/M	N/M	2/175	Special test - ran 15 sec "on" cycles.

NOTES:

* Testing Completed: The first number calls out the test point completed -or- the test point during which failure occurred. The second number number is the total time, in seconds, run at that test point.

** Wear: Wear measurements are meaningful only on specimens that completed the test. Failed specimens had surface distress or material transfer and the wear data shown is for the last test point completed successfully.

Parentheses ( ) around a negative number indicates the part actually increased its thickness (buildup) during the test--due to material transfer or grooving.

N/C indicates "no change."

N/M indicates "not measured."

*** Torque: The torque value shown is torque reading taken at the end of the last point completed successfully.

TABLE 6. REVISED TEST CONDITIONS, INITIAL SCREENING TESTS					
Test Point	Unit Load (psi)	Rubbing Velocity (fpm)	PV	"On" Cycle Time (sec)	Total Test Time (min)
1	1000	100	100,000	5	2
2	3000	145	435,000	5	5
3	6000	190	1.14×10^6	5	10

The selected material combinations and lubricants are shown in matrix form in Figure 12. The test results are also summarized in Table 5.

A study of Table 5 shows the following test results:

- Both the nonmetallic nut materials, P-3310 carbon graphite and P-59-L leaded graphite, showed promise of operating at the most severe loading conditions. Beryllium copper appeared to be the best of the metallic nut materials. No decisive trends could be observed relative to the various screw materials.

- Specimens lubricated with "Vitrolube" always achieved operation at the third test point. Time of operation at the third test point ranged from 10 seconds to the full planned test time of 600 seconds.

- The majority of specimens lubricated with "Electrofilm 5306" achieved test operation at the third test point. Time of operation at the third test point ranged from 10 to 405 seconds.

- Of the twelve grease-lubricated test specimens tested, only three specimens made it to the third test point. The time of operation at the third test point ranged from 35 to 105 seconds. The MIL-G-83261 grease may have been a bit more effective at test conditions than the MIL-G-83363 grease.

- Of the five completely nonlubricated material pairs, only one made it to test point three (P-59-L versus Nitralloy).

- The material combination originally chosen for the screw and nut, represented by Test No. 31, achieved load test point 3, but for only 10 seconds, confirming that this combination was significantly less desirable than others in the test matrix.

Additional Phase I Testing

In order to help differentiate the test materials on a more clear-cut basis, a more rigorous set of conditions was developed for an additional material selection test. The "On" cycle time for test points 1, 2, and 3, Table 6, were changed to 15, 15, and 10 seconds respectively, and the total test times were 2, 5, and 40 minutes respectively.

The additional material selection tests (39 through 56, Table 7) were run in a selective sequence (rather than conducting a full test matrix). Tests were run as follows:

- The four best candidate nut materials were each run against chrome-plated 17-4PH screw specimens (A1-1784-8B) plated with "Vitrolube."

- P-3310 and Berylco-25 nut materials, which proved to be the best two of the four tested, were then each run against screw specimens fabricated in VASCO 300 (8C) and Nitralloy (8D).

The results of the test series, also summarized in Table 7, are as follows:

P-3310 ran best with VASCO 300 (the only combination to achieve the planned test time of 40 minutes at the highest load), and second best with 17-4PH (chrome-plated).

Berylco-25 ran best with Nitralloy and second best with 17-4PH (chrome-plated).

Nitralloy 135-M was dropped from the test program at this point because of serious questions about its availability in the size, shape and quality required for a full-scale jackscrew.

TABLE 7. SUMMARY OF ADDITIONAL SCREENING TESTS.

Test No.	Nut P/N	Screw P/N	Lubricant	Testing Completed* (pt-sec)	Wear (.001 in.)** Nut	Wear (.001 in.)** Screw	Torque*** (pt/in.-lb)	Remarks
39	10A	8B	Vitrolube	3-1740	12	9	3/435	
40	10B	8B	Vitrolube	3-114	7	30	3/550	
41	10D	8B	Vitrolube	3-1210	4	12	3/250	
42	10F	8B	Vitrolube	3-27	6	14	3/750	
43	10A	8D	Vitrolube	3-680	11	6	3/240	
44	10A	8C	Vitrolube	3-2400	5	10	3/225	
45	10D	8D	Vitrolube	3-1845	6	6	3/325	
46	10D	8C	Vitrolube	3-550	9	1	2/150	Screw sample also used in test 44.
47	10D	8C	Vitrolube	3-1310	3	3	3/185	
48	13C	8B	Vitrolube	3-10	N/M	22	2/200	Molalloy sample heat shrunk into steel casing.
49	13C	8C	None	2-176	11	4	1/<25	Ditto.
50	13C	8C	None	1-308	N/M	N/M	1/20	Ditto. (Repeat)
51	13C	8C	None	1-300	N/M	N/M	1/25	Ditto. (Repeat)
52	10C	8D	None	6000 psi	2	1	350	Special test @ 90 fpm.
53	10C	8D	None	6000 psi	1	N/C	325	Special test @ 100 fpm.
54	10C	8D	None	6000 psi	1	1	325	Special test @ 120 fpm.
55	10C	8D	None	6000 psi	N/M	N/C	375	Special test @ 90 fpm.
55-1	10C	8D	None	6000 psi	2	N/C	275	Special test @ 145 fpm.
56	10C	8D	None	6000 psi	1	N/C	235	Special test @ 190 fpm.

NOTES:

* Testing Completed: The first number calls out the test point completed -or- the test point during which failure occurred. The second number is the total time, in seconds, run at that test point.

** Wear: Wear measurements are meaningful only on specimens that completed the test. Failed specimens had surface distress or material transfer and the wear data shown is for the last test point completed successfully.

N/C indicates "no change."

N/M indicates "not measured."

*** Torque: The torque value shown is torque reading taken at the end of the last point completed successfully.

Phase II Testing

A second-phase ring test was conducted to obtain a better statistical base on the most promising configuration, to conduct overload tests and to make time histories of torque and temperatures. In order to achieve the more stringent test conditions of this phase and to obtain more useful test data, the MAIC test rig was modified as follows:

- A Piezo-electric load cell was installed in the test rig torque arm and electrically connected to a precision oscillograph to monitor and record torque.
- The base of the test rig was reinforced to stiffen it and permit testing to 10,000 psi unit loading.
- More thermocouples were added to the nut and screw material test specimens, and other corrections were made to permit the simultaneous recording of the following data points: six nut temperatures, one screw temperature, and one torque reading.

Phase II test runs (101 through 118, Table 8), were conducted as follows:

- Two tests were conducted for each possible combination of the following materials:

Screw Specimens	Nut Specimens
8B (17-4PH stainless steel with chrome plate)	10A (carbon graphite)
8C (maraging steel)	10D (beryllium copper)
8E (maraging steel with chrome plate)	13C (Molalloy)

(Note: Vitrolube lubricant was used on all screw specimens and on the beryllium-copper nut specimens.)

It should be noted that the Molalloy was included as a candidate nut material even though it did not fare particularly well at the higher load levels in the initial screening tests. The reason for this continuation was an indication that very low friction levels might be achieved with Molalloy if the tendency for material breakup could be avoided.

The test results for Phase II testing are summarized in Table 8 and in Figure 14. Figure 14 illustrates in bar-graph form the test times accumulated by the various material combinations at the specified load levels. As can be seen, the P-3310 carbon graphite was clearly the preferred nut material choice, achieving the maximum scheduled test time of 40 minutes at the third load level in six out of eight tests. The maraging-steel/carbon-graphite combination achieved the full scheduled test time for each of the three tests conducted.

The second best nut material was beryllium copper, reaching the third load level with all eight specimens, but not achieving the scheduled test time of 40 minutes with any of them. The chrome-plated 17-4PH stainless-steel screw specimens gave the most consistent endurance times with the beryllium-copper nut.

The Molalloy nut specimens, despite the care taken to avoid edge chipping (shrink fitting into steel rings and chamfering the exposed edges), did not fare well at the highest load level, and three samples did not even survive five minutes at the second load level. For this reason, Molalloy was eliminated from further consideration.

Overload Tests

After completion of the tests at the third load level, the two preferred combinations - Vascomax-300 maraging steel with P-3310 carbon graphite and chrome-plated 17-4PH stainless steel with Berylco-25 beryllium copper, with Vitrolube dry-film lubricant - were tested at overload conditions to verify suitability in case of higher-than-expected nonuniform load distributions in the actual screw threads (test runs 122, 123, 126, and 127, Table 8). In this test sequence, pressures of 8000 and 10,000 psi were applied to the test specimens rather than the previous test maximum of 6000 psi. The overload test conditions are shown in Table 9.

TABLE 8. SUMMARY OF FINAL MATERIAL SELECTION TESTS.

Test No.	Nut P/N	Screw P/N	Lubricant	Testing Completed* (pt-sec)	Wear (.001 in.)** Nut	Wear (.001 in.)** Screw	Torque*** (pt/in.-lb)	Remarks
101	10A	8B	Vitrolube	3-2400	11	8	3/350	
102	10A	8B	Vitrolube	3-2400	7	10	3/350	
103	10D	8B	Vitrolube	3-810	3	15	2/110	
104	10D	8C	Vitrolube	3-15	8	6	2/100	
105	10D	8C	Vitrolube	3-35	2	5	2/100	
106	10D	8B	Vitrolube	3-700	10	9	2/125	
107	13C	8B	Vitrolube	3-20	N/M	14	2/85	
108	13C	8B	Vitrolube	3-20	3	14	2/90	
109	10A	8C	Vitrolube	3-2400	10	5	3/370	
110	10A	8C	Vitrolube	3-2400	5	5	3/310	
111	10D	8E	Vitrolube	3-30	6	14	2/80	
112A	10A	8E	Vitrolube	2-285	8	1	1/65	Screw ran on used side.
112B	10D	8E	Vitrolube	3-20	5	2	2/100	Copper transfer to screw.
113	13C	8C	Vitrolube	3-0	7	16	2/550	
114	13C	8E	Vitrolube	3-30	4	60	2/35	
115	10A	8E	Vitrolube	3-2400	1	7	3/360	
116	10A	8E	Vitrolube	3-460	6	13	2/130	
117	13C	8C	Vitrolube	2-45	2	17	1/30	
118	13C	8E	Vitrolube	2-105	4	7	1/30	
119	10A	8B	Vitrolube	3-470	5	7		Temperature gradient test.
120	10D	8B	Vitrolube	3-95	12	9	2/80	Temperature gradient test.
121	10D	8B	Vitrolube	3-40	12	8	2/100	Temperature gradient test.
122	10D	8B	Vitrolube	4-570	12	9	2/125	Overload test - 8000 psi.
123	10D	8B	Vitrolube	5-15	8	10	2/85	Overload test - 10,000 psi.
124	10D	8B	Vitrolube	3-490	11	11	2/120	
125	10A	8C	Vitrolube	3-2400	6	8	3/270	Temperature gradient test.
126	10A	8C	Vitrolube	5-600	5	12	5/270	Overload test - 10,000 psi.
127	10A	8C	Vitrolube	4-600	2	10	4/265	Overload test - 8000 psi.
128	10A	8C	Vitrolube	4-30	14	11	3/270	Reduced section, nut specimen. Completed normal test of 40 min at point 3.

NOTES:

* Testing Completed: The first number calls out the test point completed -or- the test point during which failure occurred. The second number is the total time, in seconds, run at that test point.

** Wear: Wear measurements are meaningful only on specimens that completed the test. Failed specimens had surface distress or material transfer and the wear data shown is for the last test point completed successfully.

N/M indicates "not measured."

*** Torque: The torque value shown is torque reading taken at the end of the last point completed successfully.

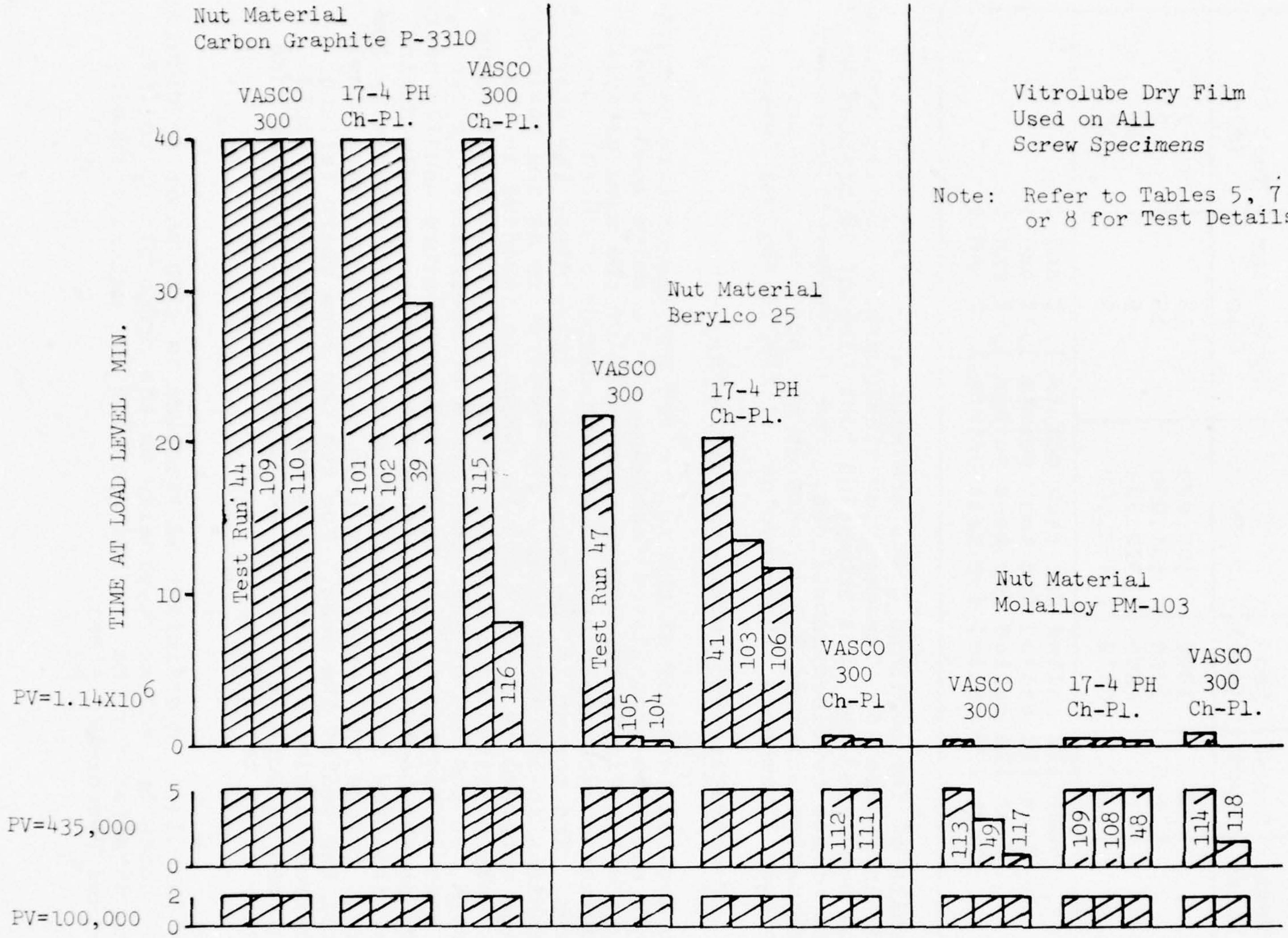


Figure 14. Summary of Ring Test Results.

TABLE 9. OVERLOAD TEST CONDITIONS.					
Test Point	Speed (rpm)	Load (psi)	PV	Cycle Time (sec)	Test Time (min)
1	100	1000	100,000	15	2
2	145	3000	435,000	15	2
4	190	8000	1,520,000	5	10
5	190	10000	1,900,000	5	10
Test 122 called for test points 1, 2 and 4 Test 123 called for test points 1, 2 and 5 Test 126 called for test points 1, 2 and 5 Test 127 called for test points 1, 2 and 4					

Results of the overload tests are also summarized in Table 8 and in Figure 15. The maraging-steel/carbon-graphite combination completed the full scheduled test time of 10 minutes at each of the overload conditions. The 17-4PH/beryllium-copper combination survived 9.5 minutes at the 8000-psi load level, but only 15 seconds (three cycles) at the 10,000-psi level.

Temperature and Torque Measurements

The typical behavior of the torque and nut temperature measurements over the accumulated test time at the third load level is shown in Figure 16. Three tests, all of the same material combination (chrome-plated 17-4PH and carbon graphite), are shown. The torque is the mean running value after the starting transient in any given cycle. The temperature is the maximum value recorded by a thermocouple, which is imbedded in the nut body or situated on the outer edge of the simulated screw thread, at the end of any given cycle of operation. As can be seen in Figure 16, both torque and temperature usually start high at the beginning of the test but decrease rapidly during the first few cycles. After bottoming out at a relatively low level, torque and temperature then increase again very slowly until the end of the test. For the one case where failure occurred, a similar pattern was evident except for a rapid increase in both temperature and torque in the three or four cycles prior to failure.

The calculated coefficient of friction is also shown in Figure 16. Quite low values, typically on the order of .05 to .07, were obtained for the carbon-graphite nut material running against Vitrolubed steel.

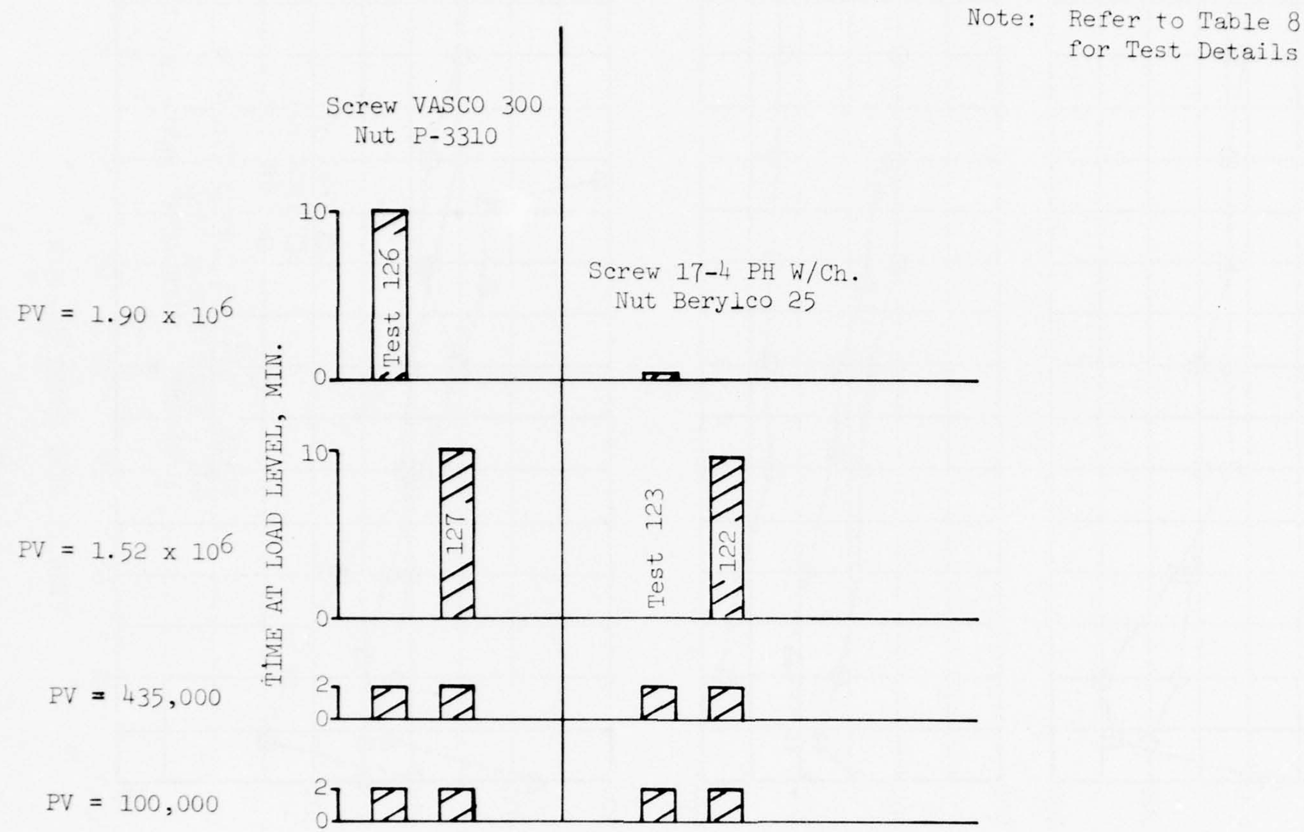


Figure 15. Summary of Overload Ring Tests.

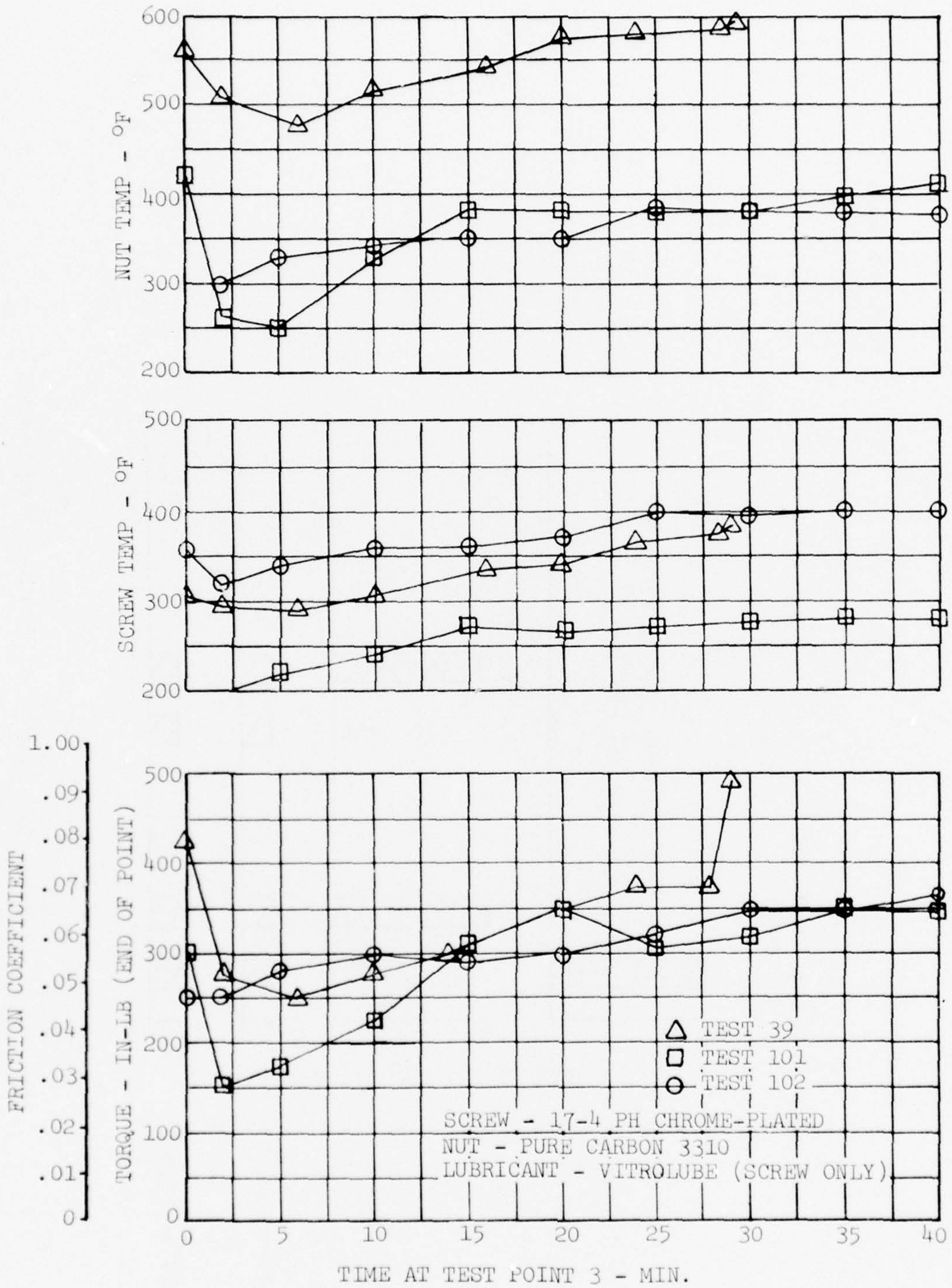


Figure 16. Summary of Ring Test Oscillograph Data.

As previously noted, the measured temperature in the ring tests should not be considered representative of the actual screw/nut application. The ring specimens rub for 10 seconds with nowhere for the heat to dissipate, whereas a real nut is continually advancing onto the relatively cold steel of the screw. For the same reason, the recorded wear rates of the ring tests are probably not representative of the actual application. However, the ring tests are expected to provide reliable information on the relative merits of the various material combinations.

Thermal Gradient and Time History Tests

A thermal gradient test was set up to map the heat flux and quantify the specific thermal gradients set up within nut specimens of the two most promising nut materials.

The thermal gradients were determined by mounting six thermocouples in the nut locations spelled out in the sketch shown on Figure 17 and running the test at the conditions described under Additional Phase I Testing, page 44.

Thermal gradient test runs 119, 120, 121, and 125 were run with the materials shown in Table 10.

TABLE 10. THERMAL GRADIENT TEST MATERIALS.

Nut	Test	Screw
P-3310	119	17-4PH (chrome-plate)
Berylco-25	120-121	17-4PH (chrome-plate)
P-3310	125	VASCO 300

All screw specimens and the beryllium-copper nut were coated with Vitrolube.

Tests 119, 120 and 121 were inconclusive due to the difficulty in keeping all six thermocouples functioning throughout the test. Therefore, extreme care was taken in setting up test 125. As a result, five of the six thermocouples produced readings throughout this run. Figure 17 is a plot of the transient temperature and torque data recorded during a typical "on" cycle of test No. 125. The unit load during this cycle was 6000 psi.

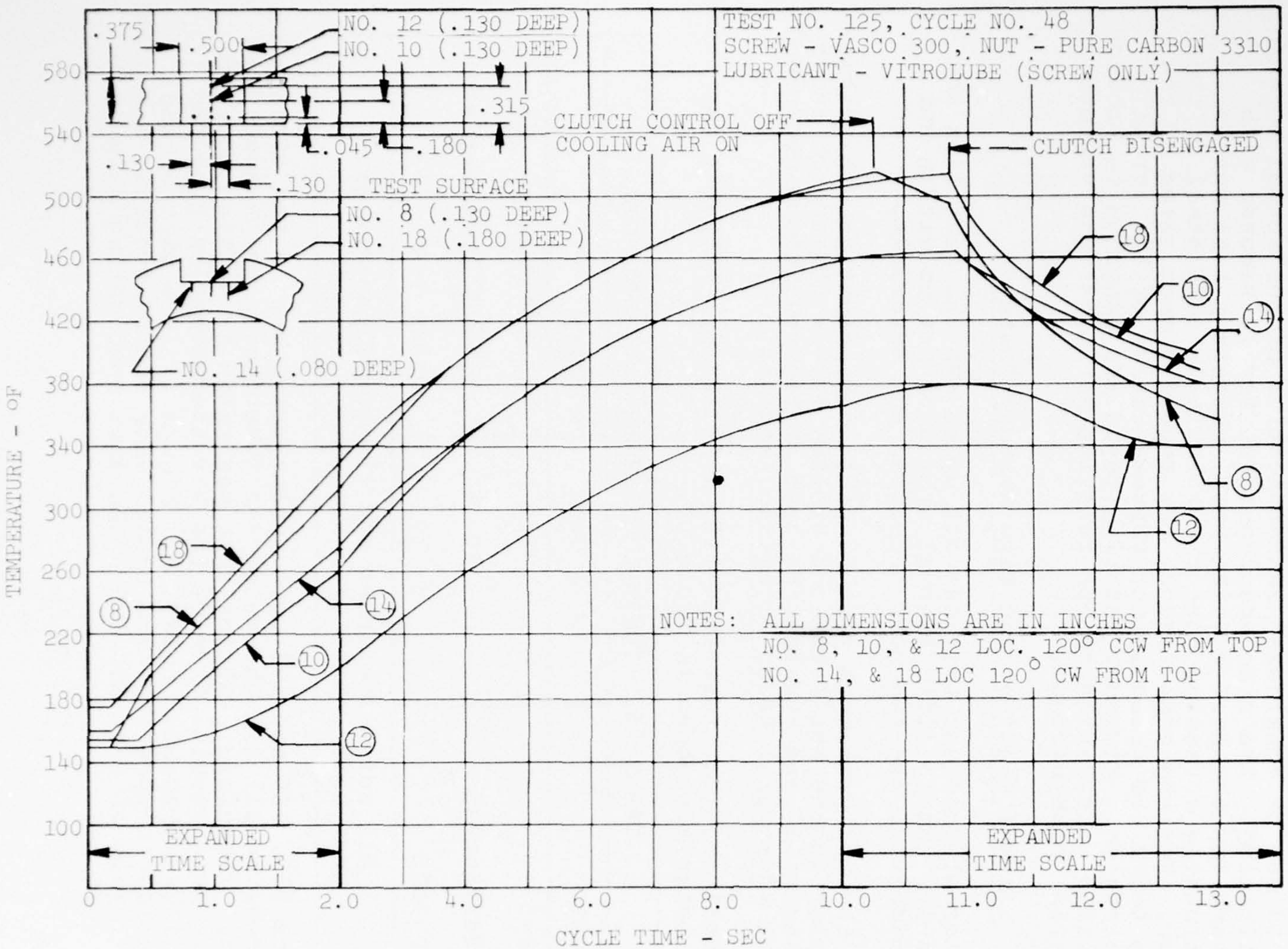


Figure 17. Results of Thermal Gradient Tests.

Referring to Figure 17, the thermal mapping of the P-3310 nut shows that, at the end of the test-point-three 10-second cycles, the ΔT from the front of the nut to the rear was $135^{o}F$. This is equivalent to a temperature gradient of $0.6^{o}F$ per .001 inch. The highest rate of change of temperature occurred at the start of the cycle; a maximum rate of approximately $80^{o}F$ per second was recorded at the thermocouple closest to the inside edge of the contact face of the nut.

Shear Strength Test of P-3310 Nut

The above materials evaluation testing was conducted with nut specimens .375 inch thick, which is a stronger section than that of the nut proposed for the actual application. To evaluate the shear strength of P-3310 in the actual nut configuration, a test specimen, nut 10A, was modified to the configuration shown in Figure 18. This test configuration produces a cross section similar to the root of the proposed nut design and simulates the stress concentrations anticipated on the back side of the thread due to the fillet in the threading on the screw.

The P-3310 nut shear test was conducted as test 128 at the conditions defined under Additional Phase I Testing, page 44. The P-3310 shear test nut completed all test points successfully. The specimen appeared to be in excellent condition.

The P-3310 shear-test nut was reinstalled in the test rig and subjected to overload testing at 8000 psi contact pressure (shear stress at root = 4200 psi). The shear-nut specimen survived five cycles of 5-second testing at the 8000-psi load level, but failed in shear at the end of the sixth test cycle.

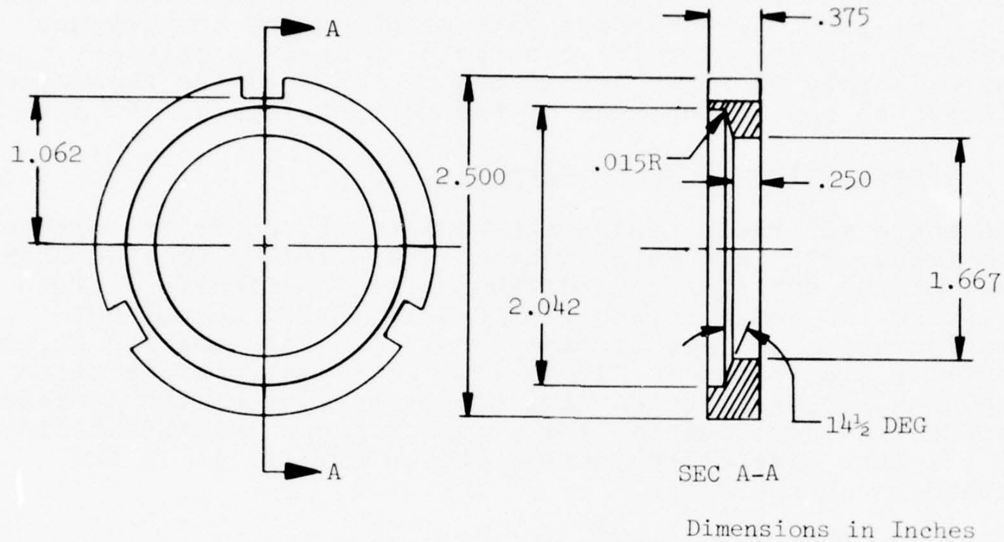


Figure 18. Reduced Section Nut Specimen.

Test Results and Conclusions

Based on the data generated during this materials test program, the following conclusions were reached.

Best Material Combinations:

	Nut		Screw
Best	P-3310	with	VASCO 300 (Vitrolube)
Second Best	Berylco-25	with	17-4PH (chrome-plate) (Vitrolube)

Both of these combinations were tested successfully at PV values well above one million, considerably above the nominal design value for the intended application. The best combination listed above survived all planned tests without a single contact surface failure when the preferred dry-film lubricant (Vitrolube) was utilized on the screw material.

Dry Film Lubricants:

Best was NPI "Vitrolube." Only "Electrofilm 5306" and "Vitrolube" dry films were tested. Both dry films will work for a period of time at high PV levels. However, the "Vitrolube" appears to perform well for longer periods of time.

MIL-G Greases:

Grease was generally unsuccessful in this application. Both greases tested had detrimental effects on the performance and life expectancy of the carbon-graphite materials.

DESIGN ANALYSIS OF FULL-SCALE COMPONENTS

Blade Spar and Torque Tube

Blade loads were calculated by means of an aeroelastic analysis developed specifically for the TRAC program. Static and vibratory blade spar and torque tube stresses were calculated for a range of flight conditions. The conditions evaluated included a range of forward speeds from 100 to 360 knots, three values of diameter, variations of lift and propulsive forces, and the effects of gusts and operation at altitude. The results of these analyses are reported in detail in Reference 5 and will not be repeated here.

Jackscrew

The threaded portion of the new jackscrew fabricated for this investigation has an outside diameter of 1.8750 inches, a pitch diameter of 1.7083 inches and a minor diameter of 1.5417 inches. The internal diameter (bore) of the jackscrew is .872 inch. The screw thread is double start and left-hand, and has three threads to an inch and a lead of 2/3 inch. The thread form is a buttress type, with a contact face angle of 14-1/2 degrees and a trailing face angle of 22-1/2 degrees. At 100-percent rotor speed (225 rpm), the jackscrew rpm is 337.5. This provides a rate of change of blade length of 3.75 inches per second and a total time for diameter change between 100 percent and 60 percent of 36 seconds. As a result of the material evaluation tests, two materials (17-4PH stainless steel and VASCO 300 maraging steel) were selected for the full-scale design.

The static design criteria is that the ultimate load (1.5 times the centrifugal force at 125-percent rpm at full blade extension) must be carried by the jackscrew. For analysis purposes, the jackscrew material was conservatively assumed to be the 17-4PH stainless steel with a minimum ultimate tensile strength of 155,000 psi. The calculated centrifugal force at the root end of the screw at 100-percent rpm and 100-percent extension is 66,900 lb (Figure 8). At full rpm and at 60-percent extension, the centrifugal load is 36,600 lb (Figure 9). The ultimate static load at the root end is $66{,}900 \times 1.5 \times (1.25)^2 = 156{,}800$ lb. The tensile area at the minimum cross section is 1.269 in.^{2} and the maximum tensile stress is $156{,}800/1.269 = 123{,}600$ psi. This results in a margin of safety of $155{,}000/123{,}600 - 1 = .25$.

During retraction and extension cycling, the tensile load at the root end of the screw fluctuates between 66,900 lb and 36,600 lb. This is equivalent to a steady load of 51,750 lb and a vibratory load of 15,100 lb. The resulting tensile stress is $40{,}800 \pm 11{,}900$ psi. Using a statistical reduction factor of .70 and a size factor of .82, the allowable vibratory stress for 10^8 cycles at a steady stress of 40,800 psi is 39,600 psi. This stress level will accommodate a stress factor of 3.3 to account for the structural discontinuity of the screw threads. This is considered adequate for the number of fatigue cycles to be run. It should be noted that the design criteria utilized for the laboratory hardware in the present investigation are different in some respects from the criteria used in the full-scale preliminary design study of Reference 5, for which a redundant jackscrew structure was assumed.

Tension Straps

Each strap is anchored in a nut by means of a .485-inch-diameter shoulder that fits in a counterbored hole in the nut. The root end of the strap, just outboard of the shoulder, has a diameter of .375 inch to fit snugly in the hole in the nut. This diameter is larger than the main body of the strap (.316 inch diameter) to permit passage of the 3/8-24 thread at the tip of the strap during assembly. A .100-inch slot is machined into the tip of the strap to provide a means of wrenching for length adjustment.

The structural design criteria established for the tension straps is that the strap package must carry the full ultimate load at 125-percent rpm with two straps failed and the remaining ten straps loaded unequally such that the highest strap load is 110 percent of the mean load of the remaining straps. At 100-percent rpm and full extension, the tensile force at the root end of the strap due to the centrifugal force of the outboard blade and the straps themselves is 54,400 lb. The ultimate load per strap is then

$$\frac{54{,}400\ (1.1)(1.5)(1.25)^2}{10} = 14{,}020 \text{ lb}$$

The strap material is Custom-455 stainless steel heat-treated to 235,000-psi ultimate tensile strength. The area at the minimum cross section is .0764 in.^{2}, and the maximum tensile stress is 183,400 psi. The margin of safety is 235,000/183,400 = .28.

During cycling, the centrifugal load felt by the straps varies between 56,600 lb and 25,800 lb. The tensile stress per strap is thus 43,740 $\pm$ 15,600 psi. The allowable vibratory working stress level at a steady stress of 44,000 is 42,500 psi. This allowable will accommodate a stress concentration factor of 2.7, which is adequate for the strap configuration and the expected number of cycles to be run.

Blade Retention Nuts

As a result of the material evaluation tests, two nut thread materials were selected for full-scale design and evaluation. The first design features a carbon-graphite (P-3310) thread insert pressed into a steel nut body with a .005-inch interference fit. For added resistance to rotation, the insert was staked with two 1/8-inch roll pins. The threads machined in the carbon graphite were left uncoated.

The second nut set is made of solid beryllium copper. The machined threads are coated with Vitrolube dry-film lubricant.

The thread details in the two nut designs were identical so that the nuts could be used interchangeably if necessary.

Six nuts, each with five threads, were provided. The centrifugal force at the nuts at 100-percent rpm and 100-percent extension is 56,600 pounds, or an average of 56,600/30 = 1887 pounds per thread. The nominal projected contact area per thread is .675 in.^{2}, so that the nominal contact pressure is 2795 psi. However, it cannot be assumed that the five threads of each nut will carry equal loads, even though the multiple nut/strap configuration provides that each nut carries an equal load. Elastic deformations in both the jackscrew and the nuts will tend to load up the inboard end threads of each nut and unload the threads toward the tip. Lapping the nuts against the screw under load will permit a more even load distribution, and this operation is recommended to accomplish better thread load sharing. Wear in service will also tend to improve thread load sharing provided that lubrication breakdown does not occur as a result of an excessive initial contact pressure. In the present analysis it is assumed that lapping will provide at least an equivalent of 60-percent thread contact under load, so that the contact pressure of the effective threads is not more than 167 percent of the nominal pressure. Thus, design contact pressure is equal to 4660 psi. The corresponding shear stress at the base of the nut thread is 2040 psi. This compares with an ultimate shear stress of approximately 8000 psi for the P-3310 carbon-graphite threads and very much higher values for the alternate nut material, beryllium copper.

Jackscrew Lubrication

The jackscrew is lubricated with Vitrolube dry film. This coating, consisting of molybdenum disulphide and other solid lubricant materials in a vitreous base, proved to be superior to the other lubricants investigated in the materials evaluation phase. It worked very well with both the carbon-graphite nut material and beryllium copper. The carbon graphite by itself is also a good solid lubricant material. These materials were tested extensively in combination at contact pressures (P) up to 6000 psi and sliding velocities (V) up to 190 feet per minute for a "PV" value of 1.14 million. A limited number of overload tests were also conducted up to 10,000 psi contact pressure and up to PV values of 1.9 million. In the present design, the sliding velocity at the pitch diameter of the screw is 152 feet per minute at 100-percent rpm (225 rotor shaft rpm, corresponding to a jackscrew rotational speed of 337.5 rpm). With a calculated contact pressure of 4660 psi, the nominal PV for the present design is 708,000, substantially less than the maximum test values.

FABRICATION

No significant manufacturing difficulties were experienced in the fabrication of the blade spar, torque tube or blade retention straps. The blade spar is a standard heavy-wall, 6061-T6 aluminum extrusion with close inside dimension tolerances. The spar's outside contour was finish-machined and polished at Sikorsky Aircraft. The torque tube was drawn from 6061-T6 aluminum alloy. The blade retention straps were made from Custom-455 stainless steel.

The manufacture of the jackscrew and nuts required special consideration. For the reasons previously outlined, the final screw thread design is a double-start thread with a buttress-type thread form. The contact face angle is 14-1/2 degrees, and the trailing face angle is 22-1/2 degrees. Since the nut materials, particularly the carbon graphite, are relatively weak in shear resistance compared to the steel screw, the nut's thread thickness at the pitch diameter was biased to approximately double that of the screw. This arrangement also facilitated using the minor diameter of the thread as a close-tolerance pilot diameter to center the screw with respect to the nut during loading and cycling.

The overriding consideration in the fabrication of the jackscrew and mating nuts was to obtain, as nearly as possible, uniform contact pressure among the five nut threads. The TRAC operating conditions that tend to defeat this goal and make the load distribution among the contacting threads non-uniform are: the elastic deformations of the jackscrew and nuts, the differential thermal expansion, and the inherent manufacturing errors in the threads themselves.

In a double start thread, the manufacturing errors are of two types. An error in starting position will manifest itself as a spacing error between consecutive threads. If this is excessive, most of the load will be carried by only one thread. A lead error in one of the two threads will also show up as a pitch error between adjacent threads, although the effect of this error is limited to the length of the nut. In light of the overwhelming odds in favor of a maldistributed load on the nut threads, it was necessary to limit the spacing error between adjacent threads to .0002 inch, which required specialized equipment and precision machining. This was accomplished by the manufacturer by grinding both threads simultaneously using a double-toothed grinding wheel. Final inspection requirements were that the screws be lapped or matched to the mating threads to achieve an accumulated thread contact of at least 80 percent.

Special precautions were taken during processing to determine, for each screw material, the dimensional change incurred as a result of the heat-treat process. Once this was known, the screws could be semi-finish milled before heat treatment, saving considerable grinding time. Following the heat-treat, the outside diameters were ground to size; the drilling, reaming and boring operations were completed; and the threads were finish-ground. The bore was first gun-drilled undersize and then honed to size to eliminate the possibility of tool scratches.

Two jackscrews were fabricated. One was made from 17-4PH stainless steel heat-treated to 155,000 psi minimum ultimate tensile strength. After final grinding, the threads on this screw were coated with a thin (.0003-inch), dense coat of electrolized chrome to increase the surface hardness and improve the thread's wear resistance. This screw was matched to the carbon-graphite nut assemblies. The other screw was fabricated from VASCO 300 maraging steel hardened to 280,000 psi minimum ultimate tensile strength and left uncoated. This screw was matched with the beryllium-copper nut assemblies.

After final processing, both screws were coated with a .001-inch layer of Vitrolube dry-film lubricant in the threaded areas.

The twelve nuts were blanked out from bar stock, and the graphite inserts were shrink fitted into the six steel nut bodies. The nut assemblies were then threaded and timed so that they would have the required spacing when installed on the jackscrew. The twelve blade-retention-strap holes were then line-drilled, using special alignment fixtures and tooling to assure that the straps would pass through the clearance holes without interference or binding. The final process was the outside contouring to obtain the diamond shape characteristic of the TRAC nut's configuration.

After processing, the beryllium-copper nuts were coated with the Vitrolube dry-film lubricant. After the Vitrolube process, which involves baking temperatures to 900°F, the beryllium-copper nuts appeared to interfere on the minor diameter when run on the jackscrew and required a slight remachining to re-establish the pilot diameter.

DEVELOPMENT TESTING

TEST FACILITY

Description

The TRAC test facility is located at the South Avenue Sikorsky plant in Bridgeport, Connecticut. The prime power source for this facility is a 500-hp motor-generator set. The facility is housed in a rotor-blade storage building that is approximately 45 feet long, 9 feet wide and 8 feet high. A schematic of the test facility is shown in Figure 19.

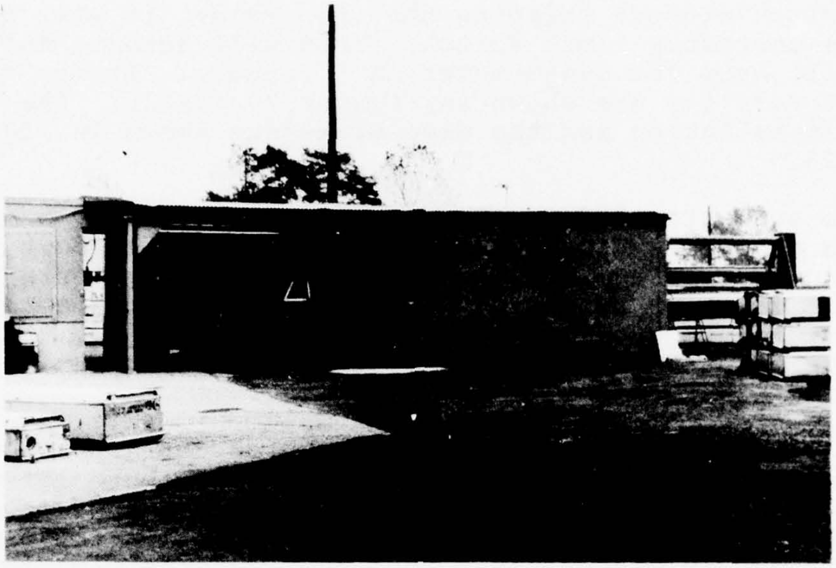
The synchronous AC drive motor has a dual function in the TRAC configuration. It mechanically powers the main hydraulic pump, delivering hydraulic fluid under pressure to the inlet port of the 40-square-inch by 144-inch-stroke hydraulic cylinder, which is used to apply simulated centrifugal loads to the blade. An auxiliary boost pump is provided to prime the main pump. Before the fluid enters the cylinder on the extension stroke, it passes through an electrically actuated regulating valve that controls and varies the cylinder pressure with blade position, thus simulating the variation in centrifugal force with rotor diameter change. On the retract stroke, the fluid is returned to the reservoir through a control valve that is similarly restricted to control back pressure. The cylinder is vented to the atmosphere on the outboard side of the piston.

The other function of the AC motor is to drive a DC generator, which in turn supplies electrical power to a variable speed dynamometer. The dynamometer drives the jackscrew through an instrumented driveshaft (to measure drive torque) at the desired screw rpm.

Interposed between the dynamometer and the torque shaft is a hydraulically actuated disc brake that is provided for emergency shutdown to prevent damage to the test components.

To simulate the aircraft configuration and to isolate the flapping blade masses from the support structure, a universal joint is provided at both the inboard blade connection and the outboard connection to the hydraulic ram.

Two sets of limit switches are provided to stop the telescoping blade at the end of its travel. The inboard switch, when tripped, relays a zero-speed command to the dynamometer, which will then come to a stop in approximately 5 seconds. This is the normal stopping procedure before reversing the screw rotation. The outboard limit switches are emergency backup switches. If the nut assembly overtravels the normal stop



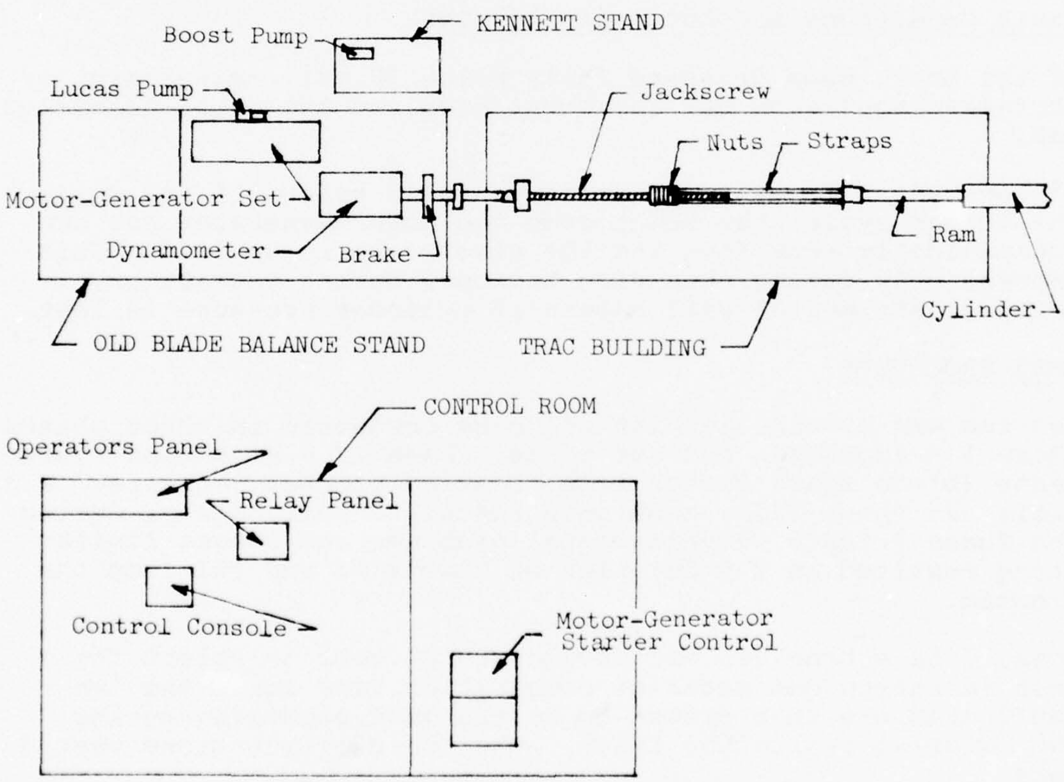


Figure 19. Full-Scale TRAC Test Facility.

limit switch without tripping the stop relay, it will contact the emergency limit switch. This will actuate the brake, which will stop the dynamometer in 5 seconds. Photos of the Test Facility are shown in Figures 20 and 21. The limit switch installation and the disc brake are shown in Figures 22 and 23.

Upon the suggestion of the Eustis Project Engineer, a movable carriage (shown in Figure 21) was installed to support the traveling nut package during the initial cycling tests and to react the friction torque developed. This arrangement replaced the dummy torque tube arrangement, and afforded the test engineers an unobstructed view of and easy access to the screw threads during cycling.

Note in Figure 20 that the outboard end of the jackscrew is soft-mounted to the facility frame to provide both lateral and vertical flexibility so that, when the carriage moves outboard, the screw can seek a centralized position relative to the nut assembly without introducing extraneous side loads.

Fault Conditions and Automatic Backups

If the boost pump pressure falls below 20 psi, relays are energized that shut off the boost pump and the motor-generator set.

If the hydraulic cylinder pressure falls below 300 psi during the extend cycle, the boost pump and motor-generator set are automatically shut down and the disc brake is applied. This happens only during extending because, during retraction, nothing detrimental will happen if cylinder pressure is lost.

TEST PROCEDURE

Testing was originally planned to be conducted in three phases: Phase I - jackscrew and nut tests; Phase II - assembled system tests (blade spar, torque tube, bearing blocks, jackscrew, and nut); and Phase III - endurance tests. Unfortunately, during the Phase I tests, developmental problems and budget limitations resulted in the deletion of Phases II and III from the program.

Phase I is a break-in and developmental test to select the best jackscrew/nut material combination from among the two configurations that proved to be the most promising during the material evaluation tests. The two configurations tested were:

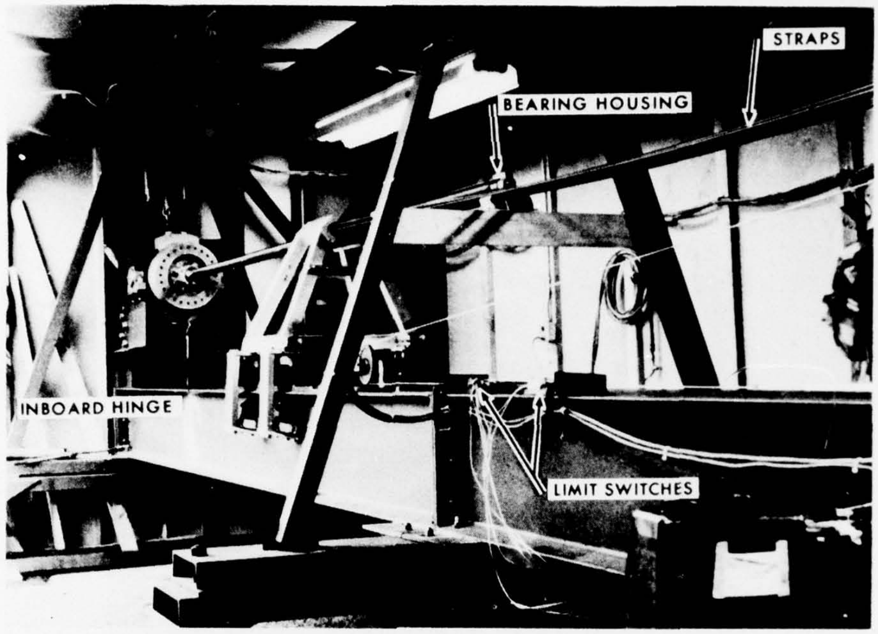


Figure 20. TRAC Test Facility - View Looking Inboard

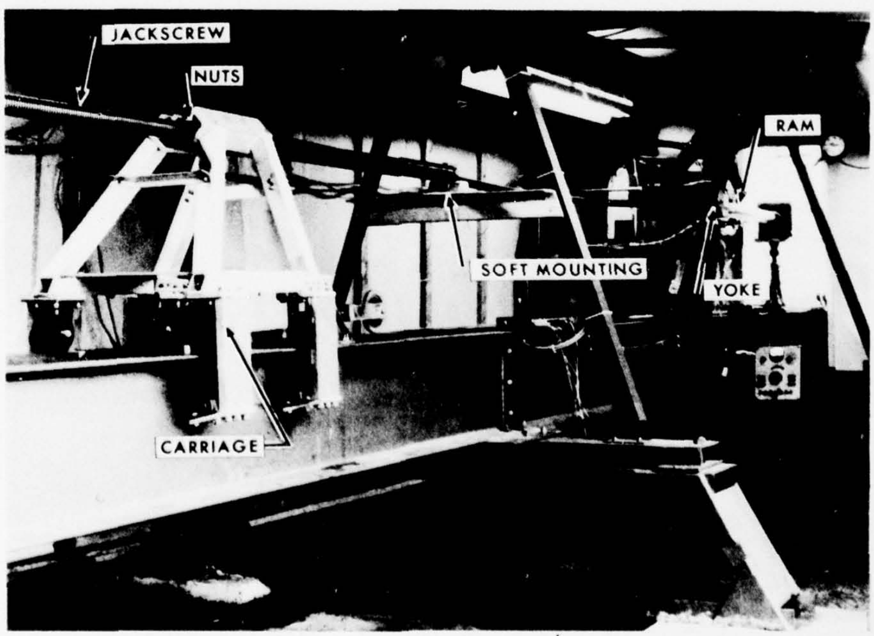


Figure 21. TRAC Test Facility - View Looking Outboard.

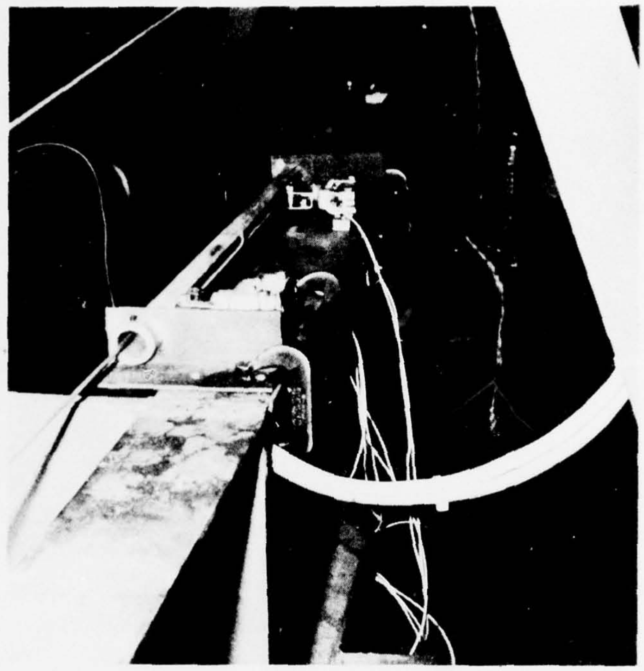


Figure 22. Limit Switch Installation.

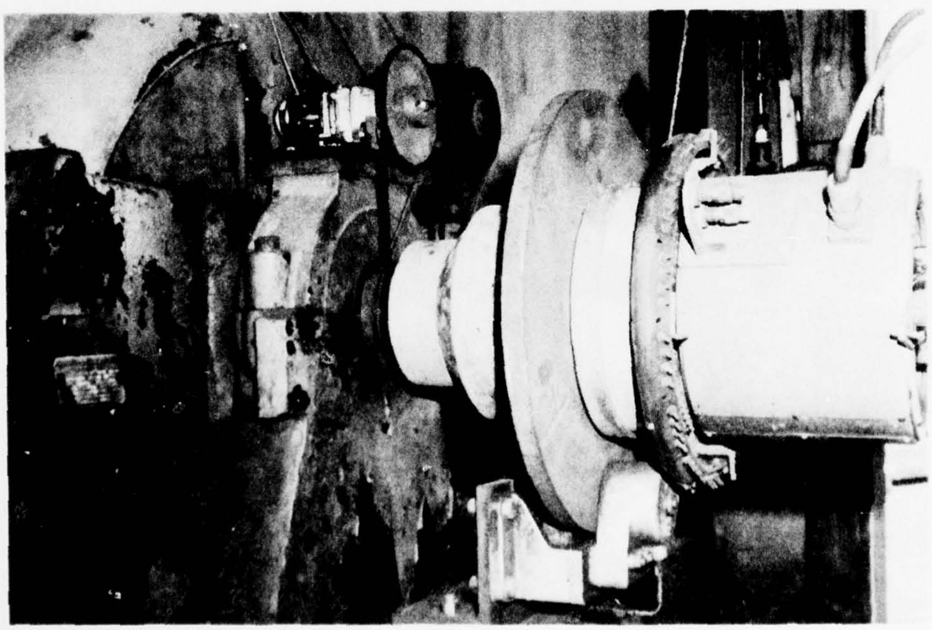


Figure 23. Disc Brake Installation.

- Maraging-steel jackscrew coated with Vitrolube dry-film lubricant running against a carbon-graphite thread insert pressed into a steel nut body.

- Chrome-plated stainless steel jackscrew running against a beryllium copper nut, both coated with Vitrolube.

The jackscrew-nut operating conditions to be simulated during this test were:

RPM (Jackscrew)	=	337 max
Centrifugal Loads	=	26,000 lb retracted to 57,000 lb extended

Up to 400 retraction/extension cycles were to be conducted on each candidate configuration unless limited by excessive temperature, friction or wear. Jackscrew and nut temperatures, wear rates and friction torque were to be the criteria for the selection of the best material combination. The test plan for this phase is summarized in Table 11.

Measurements

The test components and facility were instrumented as follows. Nut temperature was measured with a thermocouple mounted to each nut segment. Screw temperature was measured by a thermocouple contacting the face of the thread on the contact side with the sensor imbedded in a rubbing block such that the end of the thermocouple was flush with the surface. This arrangement is shown in Figure 24. The driveshaft between the dynamometer and the jackscrew was strain-gaged and calibrated to measure drive torque. Applied axial load was measured by strain gaging and calibrating the loading yoke on the hydraulic ram as shown in Figure 25. Wear in the screw and the nut threads was recorded by measurements taken across a 3/16-inch ball inserted in the thread. Incremental differences in this measurement were indicative of thread wear.

Lapping Procedure

During the initial break-in cycles run with the maraging-steel jackscrew and the carbon-graphite nuts installed and loaded with the blade retention straps, it became evident that, in spite of the precautions taken during manufacture to minimize the effect of thread mismatch, equal load sharing was, in fact, not taking place. Most of the load was being taken by only one of the two screw threads, and this effect was fairly constant along the length of the screw. In view of the relatively low shear strength characteristic of the carbon-graphite material, it was felt that this condition was unacceptable and that an attempt should be made to correct

TABLE 11. RETRACTION-EXTENSION TEST CONDITIONS.

Jackscrew	Nut	Jackscrew (rpm)	Centrifugal Load (lb)	No. of Cycles
Burnish		200	Min 1,000	2
		337	1,000	2
Vitrolube	Vitrolube	20	5,000	5
coated	coated	163	13,000	5
stainless	Beryllium	168	20,000	5
steel	Copper	337	26,000	5
38004-0033 -101	38004-0035	337	26,000 retracted to 57,000 extended	380
Burnish		200	1,000	2
		337	1,000	2
Vitrolube	Carbon	20	5,000	5
coated	Graphite	168	13,000	5
maraging	insert in	168	20,000	5
steel	steel body	337	26,000	5
38004-00033 -103	38004-00038	337	26,000 retracted to 57,000 extended	380
Optimum Combination		337	26,000 retracted to 57,000 extended	80
			TOTAL RETRACT/EXTEND CYCLES	880

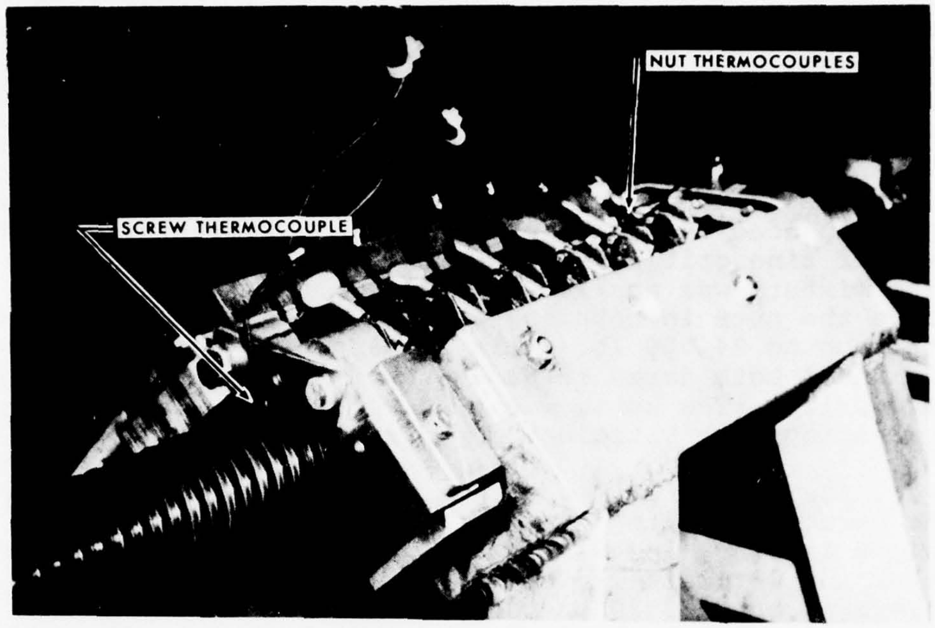


Figure 24. Thermocouple Arrangement.

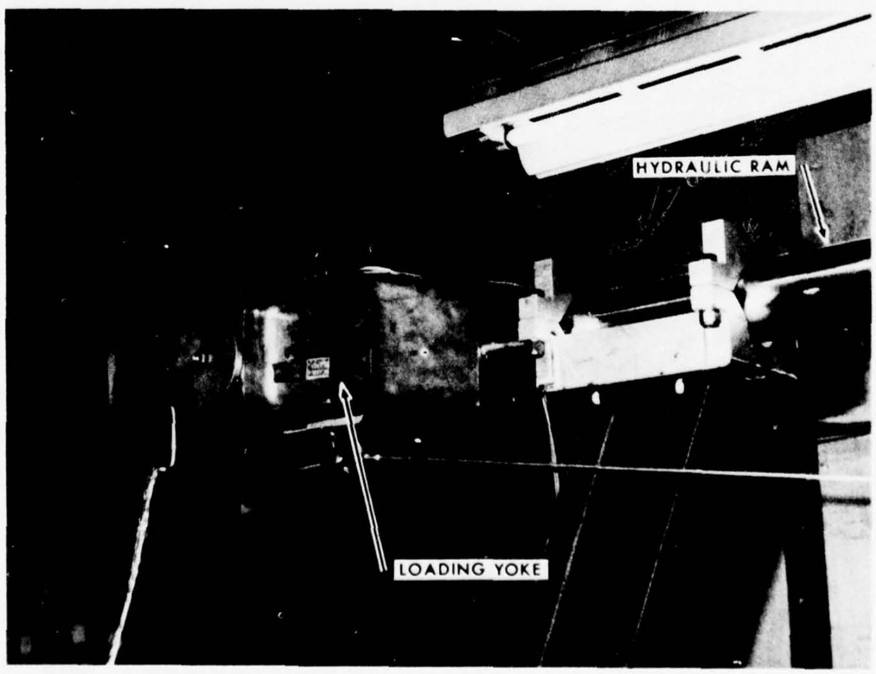


Figure 25. Axial Load Measurement Location.

the situation. This was accomplished by lapping the screw and nut together under load, thus forcing the nut to conform to the inaccuracies of the screw.

Therefore, prior to running the Phase I cycle tests, the nuts were lapped, both singly and in combination with the other five, to the jackscrew, starting with a coarse grit lapping powder suspended in Crane vehicle oil and finishing with a mixture of fine grit powder and lubriplate grease. The lapping mixture was applied manually on the turning screw as it drove the nuts in both the retract and extend directions at loads up to 24,000 lb. This lapping procedure was continued until both screw threads showed signs of sharing the loads equally. The screw was then returned to the manufacturer for recoating with Vitrolube dry film.

However, the above techniques caused fairly severe pitting in the carbon-graphite threads, especially during initial high-load lapping, and ultimately the pits had to be removed by returning periodically to no-load or low-load lapping. On the average, about .020 to .030 inch was removed from the thread surfaces during the lapping procedure. It was concluded that, although the spacing and lead errors induced by manufacturing tolerances could be corrected by lapping in the manner described, the necessity to resort to low-load lapping to remove pitting limited the ability of the lapping process to correct for any maldistribution of pressure due to the elastic deflections under load. A refined method must be found to accomplish the lapping without removing so much material.

No attempt was made to lap the beryllium-copper nuts to the stainless-steel screw at this time since it was felt that this stronger combination could tolerate the maldistribution of load.

TEST RESULTS

Testing - Beryllium-Copper Nuts

The beryllium-copper nuts were tested in combination with the stainless-steel screw in accordance with the test conditions outlined in Table 11. The screw was chrome-plated, and both screw and nuts were coated with Vitrolube dry-film lubricant. Half of a test cycle was spent retracting, half extending. All testing was conducted one-half cycle at a time, and the test specimens were allowed to cool to 100°F above ambient temperature after each half cycle. The loads were applied to the nuts through the twelve blade retention straps. Each strap length was adjusted individually by screwing it into or out of the tip block assembly attached to the end of the hydraulic ram and then locking it in place with a locknut. The straps were adjusted manually to obtain equal tension in each strap.

Initial assembly of the beryllium nuts on the chrome-plated screw revealed a distinct difference in the tightness of each nut on the screw thread. The tightness of any given nut also varied as the nut was run by hand along the length of the screw, indicating variations in the screw's plating dimensions.

The operation of the retraction mechanism, with all nuts installed, at 20 rpm and the minimum (1500-lb) load resulted in the "shaving" of particles of Vitrolube from the screw or nuts, although in most areas, including the pilot diameter, the Vitrolube assumed a polished appearance. Contact was also indicated between the nut thread and the root of the screw thread, as evidenced by Vitrolube removal and polishing of the metal screw. At higher loads, the root contact was more apparent as the polished area became wider. On several threads, copper deposits were evident on the screw at the roots of the loaded faces of the teeth. This contact was probably due to some accumulation of chrome in the filleted areas.

The Vitrolube lubricant began to disappear from the nuts upon initial testing and soon was completely removed. Galling of the nut's loaded thread face became evident as the load was increased to 13,000 pounds at 168 rpm. Squealing was evident during the extend stroke that became very loud. When the load was increased to 20,000 pounds at 182 rpm, the noise became very loud in both directions, and the Vitrolube polish disappeared from the screw threads, which then appeared dull and roughened. Two sections of screw thread had evidence of copper deposits from the galling of the nuts.

Simultaneously with the squealing during extension, the recorded steady torque, with a superimposed vibratory torque, increased drastically. Instead of increasing load, as specified in step 3 of the test program, the screw speed was increased to 337 rpm while the load was maintained at 20,000 pounds. At the higher rpm, the vibratory torque decreased, as did the audible screeching during the extension stroke. The steady torque also decreased during extension from a maximum of 8000 in.-lb at 168 rpm with a 13,000-lb centrifugal load, while the retraction torque increased with screw speed and centrifugal load. During retraction at 337 rpm with a 20,000-lb centrifugal load, smoke was seen coming from the nuts near the end of the retraction stroke. The temperature on nut 4, which had consistently been the hottest, rose to 340°F. At 337 rpm, the nut travelled onto the previously unloaded end portion of the screw. As a result of this overtravel at high speed rotation, the Vitrolube became polished with very little shaving or deformation. The copper transfer from the nuts to the screw, observed earlier, was evident on alternate teeth (single thread contact) over most of the screw after the 20,000-lb, 337-rpm run.

Removal of the nuts revealed:

(a) galling of the loaded thread face, especially on the outboard end of the nuts

(b) polishing and occasional galling of the tips of the teeth on the loaded face was further evidence of the contact at the root radius of the screw thread

(c) all Vitrolube lubricant had been removed from the loaded thread face of the nuts.

It was tentatively concluded that lapping the beryllium copper nuts against the screw, under load, would improve the loading distribution and substantially increase the probability that the system would function as desired. However, this would require not only the lapping operation but also the re-chrome-plating of the screw and reapplication of the Vitrolube dry film. Because of the shortage of funds, it was decided to terminate the testing of this configuration and proceed to the primary configuration (maraging-steel and carbon-graphite nuts).

Testing - Carbon-Graphite Nuts

The carbon-graphite nuts were tested in combination with the maraging-steel jackscrew. This combination had been lapped together as previously described. After lapping, the jackscrew threads were recoated with fresh Vitrolube dry-film lubricant.

Prior to running the planned test program, approximately ten retraction/extension cycles were run at minimum-load (1000 lb), full-speed conditions to burnish the Vitrolube contact surfaces. This polishing operation was considered necessary to prepare the Vitrolube surface for full-load operation.

The retraction-extension cycle testing was run at the test conditions of Table 12 except that, based upon the results of the test on the beryllium-copper nuts, the speed was held constant at 337 rpm.

During the cycle testing, the following interesting trends were observed:

- The squealing on the extension portion of the cycle, observed during the beryllium-copper nut tests, was still present, although measureably abated. As the test progressed, the extension-cycle noise eventually disappeared. The retraction portion was entirely smooth and quiet throughout the test.

- The coefficient of friction gradually decreased as the number of cycles increased, indicating a possible need for a longer break-in period for the Vitrolube lubricant. The coefficient of friction on the extension portion of the cycle was higher than on the retraction portion. This may be a clue to the noise phenomenon.

- After the initial break-in period, the wear rate of the carbon-graphite nuts leveled off to a fairly constant value that is considered entirely satisfactory for the intended application.

- Screw and nut temperatures were not considered excessive.

At the end of approximately 82 test cycles, a facility malfunction occurred that destroyed two of the five threads of the outboard nut segment. On the extension stroke, the limit switches failed to stop the screw rotation carriage in time. As a result of this, the outboard nut segment (number six nut) ran off the end of the jackscrew. When the load on that nut (approximately 9500 lb) was carried by the last two threads, the carbon threads sheared. There are two possible contributing factors to this mishap: (1) the severe cold that particular morning (<$10^{\circ}F$) may have affected the limit switch action and

(2) the friction coefficients, which have been decreasing with cycle time, may have dropped to a value (<0.120) where the load starts to drive the screw, thus extending the stopping time.

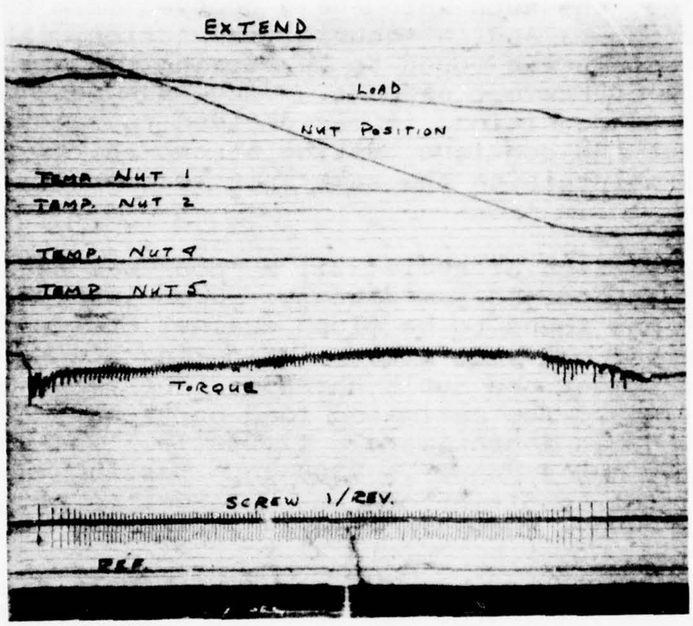
To prevent this from occurring again, the limit switches were repositioned approximately six inches further from the screw end and readjusted to give a more positive contact. In addition, changes were made in the drive motor circuitry to increase the braking characteristics of the motor when the load is driving.

Some positive results were inadvertently obtained because of the nut run-off incident. First, it confirmed that the ultimate shear strength of the carbon graphite was approximately equal to the value assumed in the design analysis and that the five threads of the nut provide a substantial strength margin at the design load. Second, it demonstrated the basic safety of the multiple-nut concept, with the full design centrifugal load being carried by the five remaining nuts after one had run off. Third, it demonstrated a capability of withstanding some degree of shock loading, because, when the threads of one nut sheared, the load carried by that nut was suddenly transferred to the remaining nuts and an elastic shock load would inevitably be felt throughout the system when the energy stored in one pair of straps is suddenly released. However, this shock load may have contributed to a decreased useful wear life for the rest of the threads.

The remainder of the test cycles were conducted using five nut segments instead of six. To maintain the same unit load, the centrifugal load was decreased by 1/6. The new load limits were then 22,000 lb when retracted and 47,000 lb when extended. Recorded data for a typical extend and retract cycle is shown in Figure 26.

After 130 total cycles, the nuts were removed from the screw for inspection. All nut segments were found to be in relatively good condition, and they were reassembled for further testing. The measured wear rate for the carbon threads at this time was running about .004 to .005 inch per 50 cycles.

Care was taken when installing the nuts on the jackscrew to ensure that: (1) the 14-1/2-degree face angle of the nut contacted the 14-1/2-degree face of the screw, and (2) the nut was restarted on the same screw thread each time.



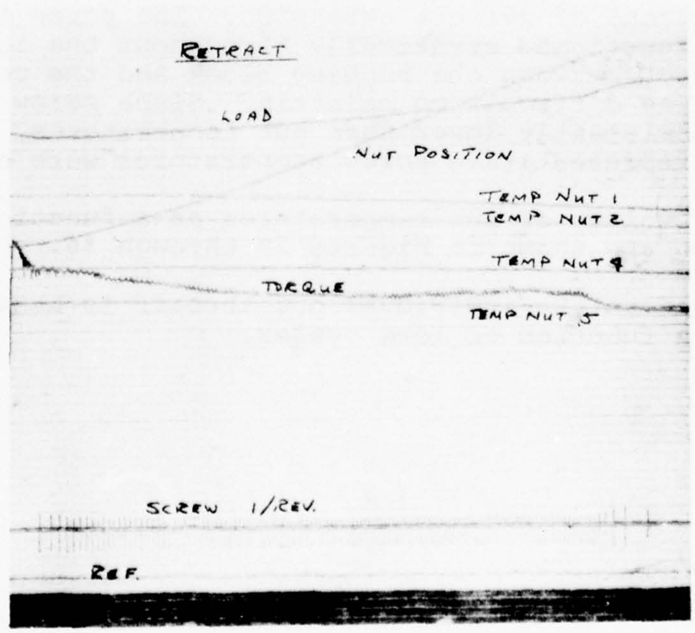


Figure 26. Typical Oscillograph Data from Retraction/Extension Test.

After 238 cycles, the nuts were again removed from the screw for wear measurements and for general inspection. At this time, the number three nut was found to be missing two carbon threads from the middle of the nut segment. Since the rest of the nuts were in good condition, it was decided to remove the number three nut and continue testing at two-thirds the original load, maintaining the same unit load on the remaining four nuts.

On the retract portion of cycle 321, a "pop" was heard and the screw was stopped immediately. Upon inspection, the number one nut was found to be flush against number two, and the rods to number one were slack, indicating a complete failure of the number one nut's threads. The remaining three nuts held the load. The estimated load on these nuts in the holding position was approximately 12,000 lb. The normal maximum load per nut segment is 9300 lb. Testing was terminated at this time. The condition of the nut segments at the completion of testing is shown in Figures 27 and 28.

Thus, the combination of the carbon-graphite nuts running with the maraging-steel jackscrew was considered to have failed after 320 test cycles. The test results for representative test cycles 54, 207, 256 and 320 are presented in Table 12. Tabulated are centrifugal load, drive torque and nut temperatures as functions of percent extension. The screw temperature thermocouple functioned erratically throughout the test and, positive contact between the rubbing block and the contact face of the screw was difficult to maintain. Since screw temperatures were consistently lower than nut temperatures, attempts at recording representative screw temperatures were abandoned.

The tabulated values of nut temperatures as a function of blade position are shown in Figures 29 through 36.

The thread wear on the individual nut threads is shown in Figure 37 as a function of load cycles.

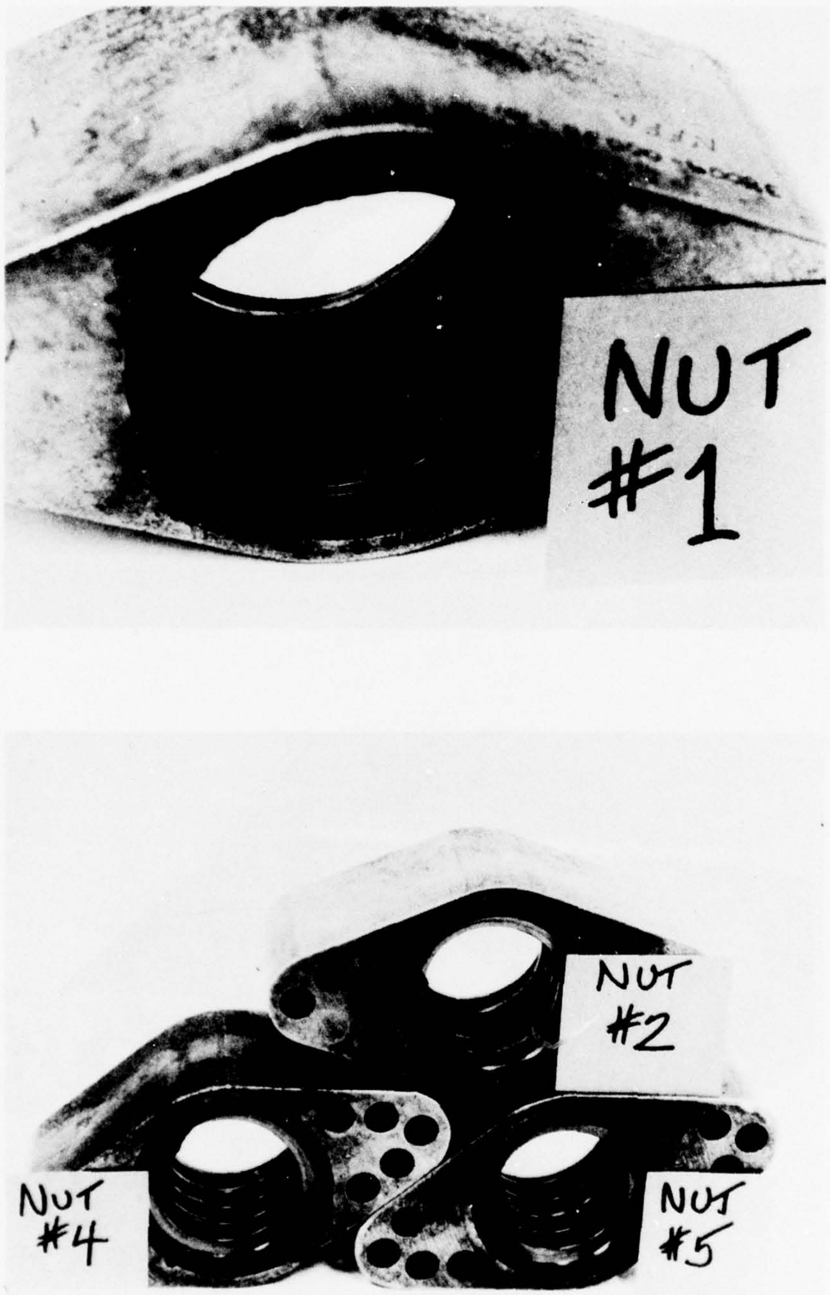


Figure 27. Carbon-Graphite Nuts at Test Conclusion.

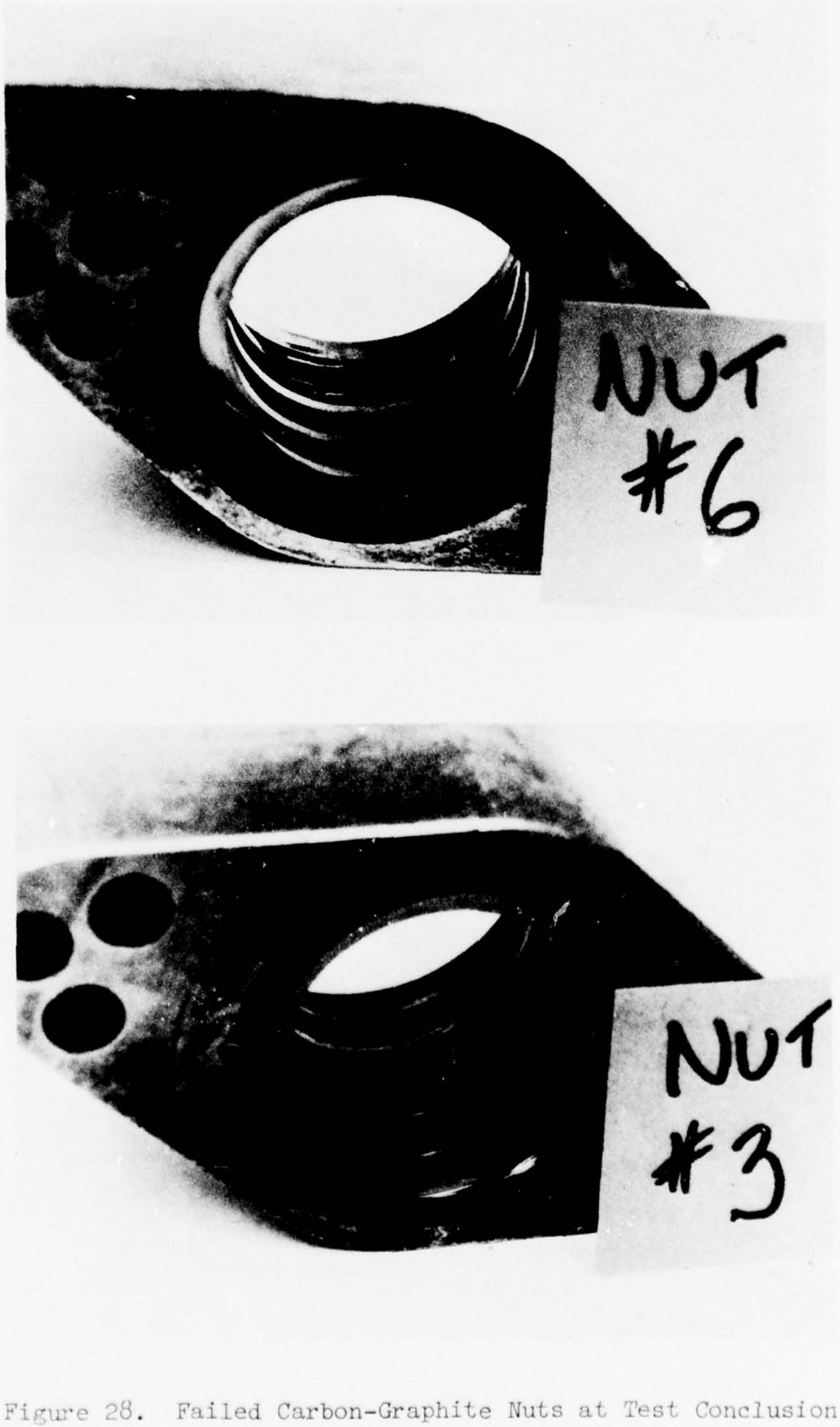


Figure 28. Failed Carbon-Graphite Nuts at Test Conclusion.

TABLE 12. TEST RESULTS - CARBON-GRAPHITE NUTS.

Cycle	Percent Extended	Nut Temperature (^{o}F) 1	2	3	4	5	6	Load (K lb)	Torque (in -lb)
53.5	0	115	115	125	130	130	125	28	-2200
	10	115	115	127	130	130	125	26	-1500
	20	120	120	130	140	140	130	29	-2000
	30	120	125	140	145	145	130	32	-2000
	40	125	130	150	155	145	140	36	-2000
	50	125	140	160	170	165	150	39	-2000
	60	135	140	175	175	165	150	42	-1900
Extend	70	140	155	180	190	175	165	45	-1500
	80	150	165	195	200	190	165	48	-2900
	90	150	170	210	220	195	175	51	-2500
	100	165	180	225	230	200	175	57	-2800
54	100	120	120	130	125	120	105	57	13000
	90	125	125	130	125	120	105	57	12000
	80	125	125	140	140	125	110	54	13000
	70	125	135	150	160	145	125	51	11100
	60	130	140	175	175	155	130	49	10500
	50	140	150	195	200	170	150	45	10400
	40	145	165	210	225	190	165	43	9900
Retract	30	150	170	225	240	200	170	40	9200
	20	160	175	250	265	215	185	36	8900
	10	170	190	265	275	240	200	34	8000
	0	175	195	275	290	245	220	28	6700
206.5	0	125	130	130	135	140		24	-1000
	10	125	135	130	145	145		22	-1000
	20	125	140	130	145	145		25	-1500
	30	130	145	135	145	145		28	-2000
	40	130	150	135	150	150		30	-1800
	50	145	165	145	150	155		32	-1500
	60	155	170	150	155	165		35	-800
Extend	70	170	180	150	160	165		38	-800
	80	170	195	160	170	175	Removed	40	-1000
	90	185	200	170	175	175		42	-1200
	100	195	225	175	180	190		46	-2500

TABLE 12. (Continued)									
		Nut Temperature (^{o}F)							
Cycle	Percent Extended	1	2	3	4	5	6	Load (K lb)	Torque (in.-lb)
207	100	150	150	145	140	130		46	13300
	90	150	150	140	140	130		47	9000
	80	150	150	140	140	130		46	7500
	70	160	160	150	145	140		43	7000
	60	165	165	150	150	145		40	6700
	50	170	175	150	150	150		38	7100
Retract	40	170	175	160	165	165		36	7900
	30	175	190	165	165	165		34	8000
	20	190	200	170	170	170		32	8200
	10	195	205	175	175	175		30	8000
	0	215	235	195	195	195		24	6500
255.5	0	160	150		150	175		20	-2000
	10	155	150		150	175		16	-200
	20	165	150		150	180		18	-200
	30	165	155		150	180		20	-200
	40	170	155		150	180		22	-100
	50	170	160		150	185		24	0
	60	175	165		160	190		27	100
Extend	70	180	165		155	190		28	200
	80	185	170		165	195	Removed	30	100
	90	195	175		165	200		32	0
	100	200	175		165	200		38	-1000
256	100	165	155	Removed	150	165		36	11000
	90	165	155		150	165		38	7000
	80	165	155		150	165		36	6000
	70	175	165		150	170		35	5900
	60	180	165		150	170		34	5900
	50	185	170		155	175		33	5900
Retract	40	190	170		160	185		30	5900
	30	200	175		160	185		28	5700
	20	205	180		170	195		26	5700
	10	220	185		170	195		25	5200
	0	230	195		175	200		18	4800

TABLE 12. (Continued)									
		Nut Temperature (^{o}F)							
Cycle	Percent Extended	1	2	3	4	5	6	Load (K lb)	Torque (in.-lb)
320	100	165	150	Removed	145	150	Removed	40	8300
	90	170	150		140	150		40	5900
	80	170	155		140	150		39	5200
	70	170	155		140	150		37	5000
Retract	60	170	160		140	160		35	4800
	50	170	165		145	165		33	4800
	40	175	170		145	165		31	4600
	30	175	170		145	165		29	4500
	20	175	170		150	165		28	4500
	10	180	170		150	170		26	4500
	0	185	175		150	170		20	3800
320.5	0	155	145		140	155		18	0
	10	160	145		140	155		17	500
	20	160	150		140	155		20	500
	30	165	150		140	155		21	900
Extend	40	170	150		140	155		25	1000
	50	170	150		140	155		26	1000
	60	170	160		140	155		27	1000
	70	175	155		140	155		29	1300
	80	180	165		145	160		31	1300
	90	185	165		145	160		33	1300
	100	190	170		145	160		38	1000

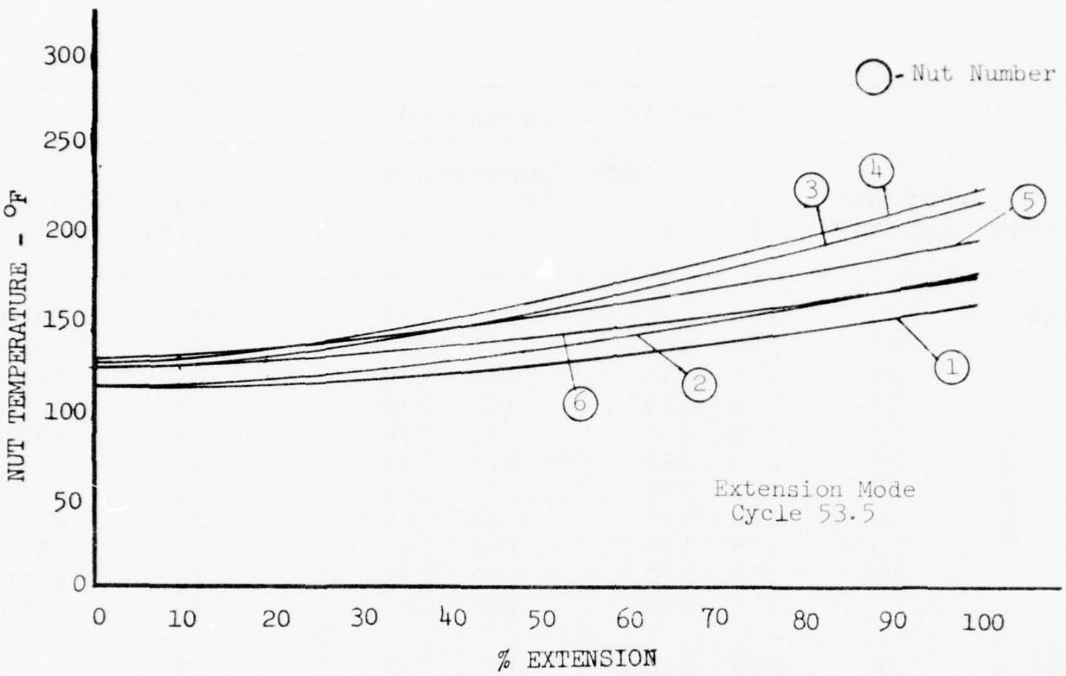


Figure 29. Nut Temperatures vs Blade Position.

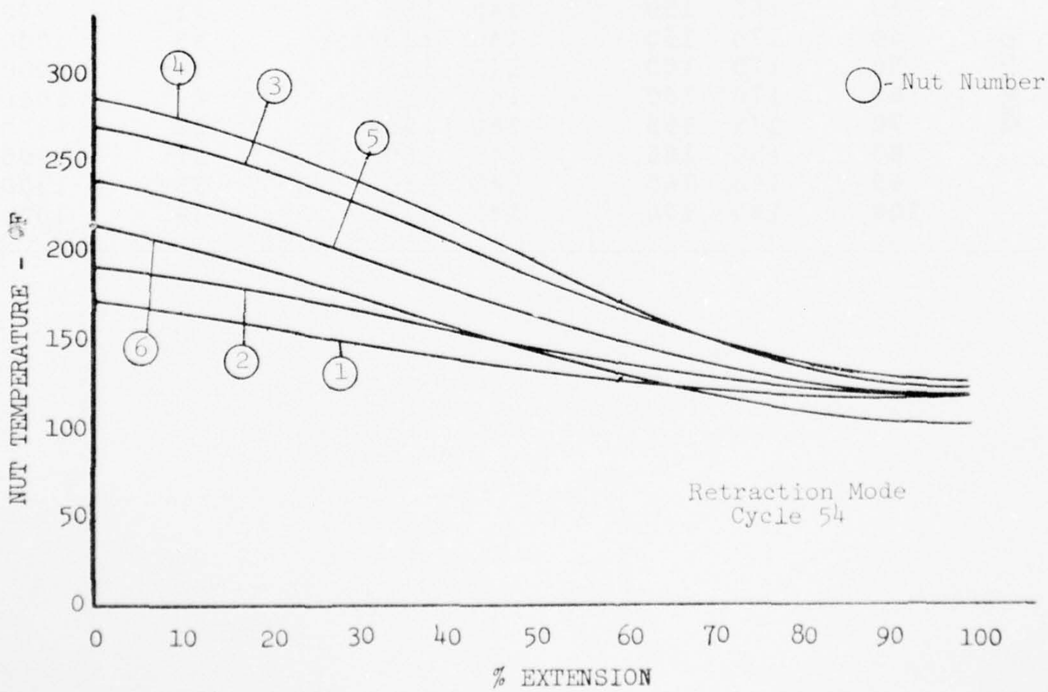


Figure 30. Nut Temperatures vs Blade Position.

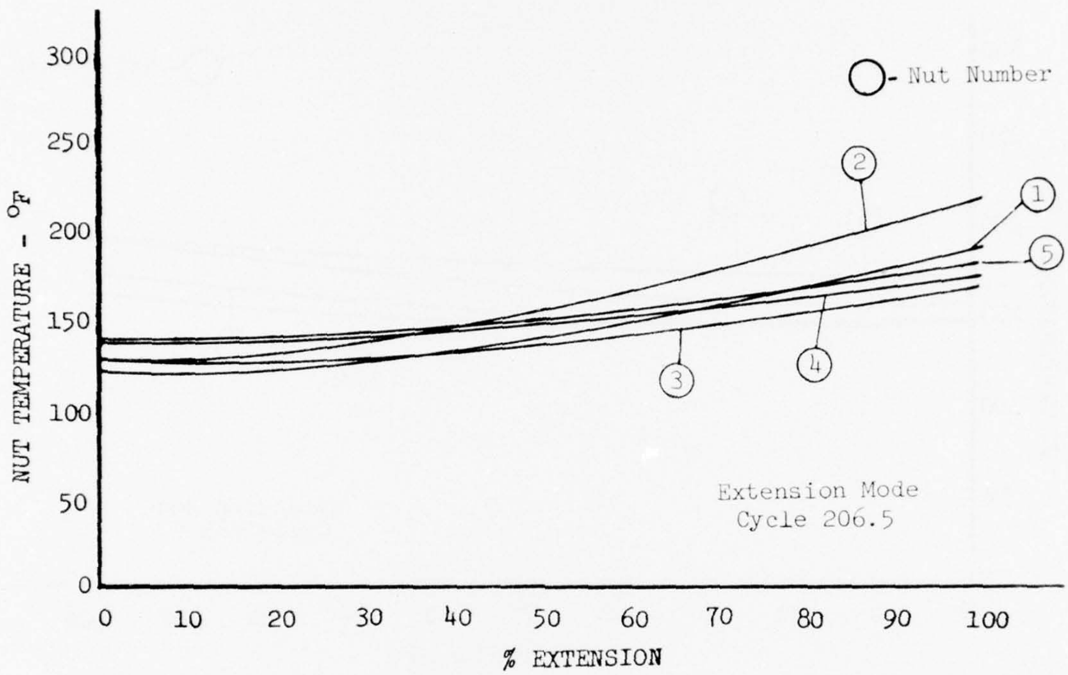


Figure 31. Nut Temperatures vs Blade Position.

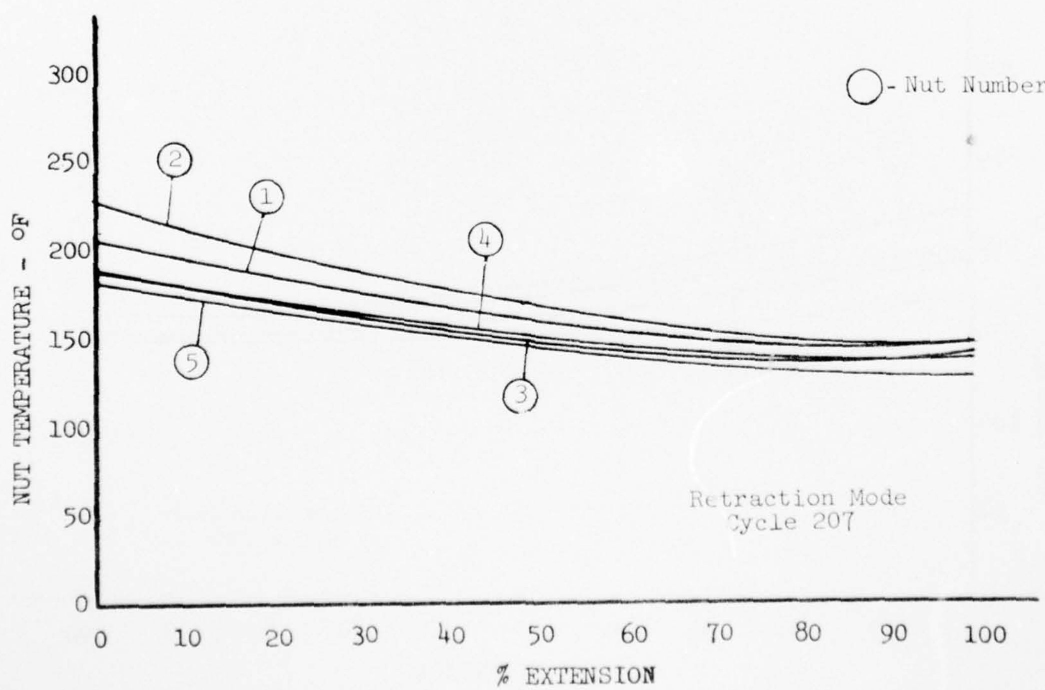


Figure 32. Nut Temperatures vs Blade Position.

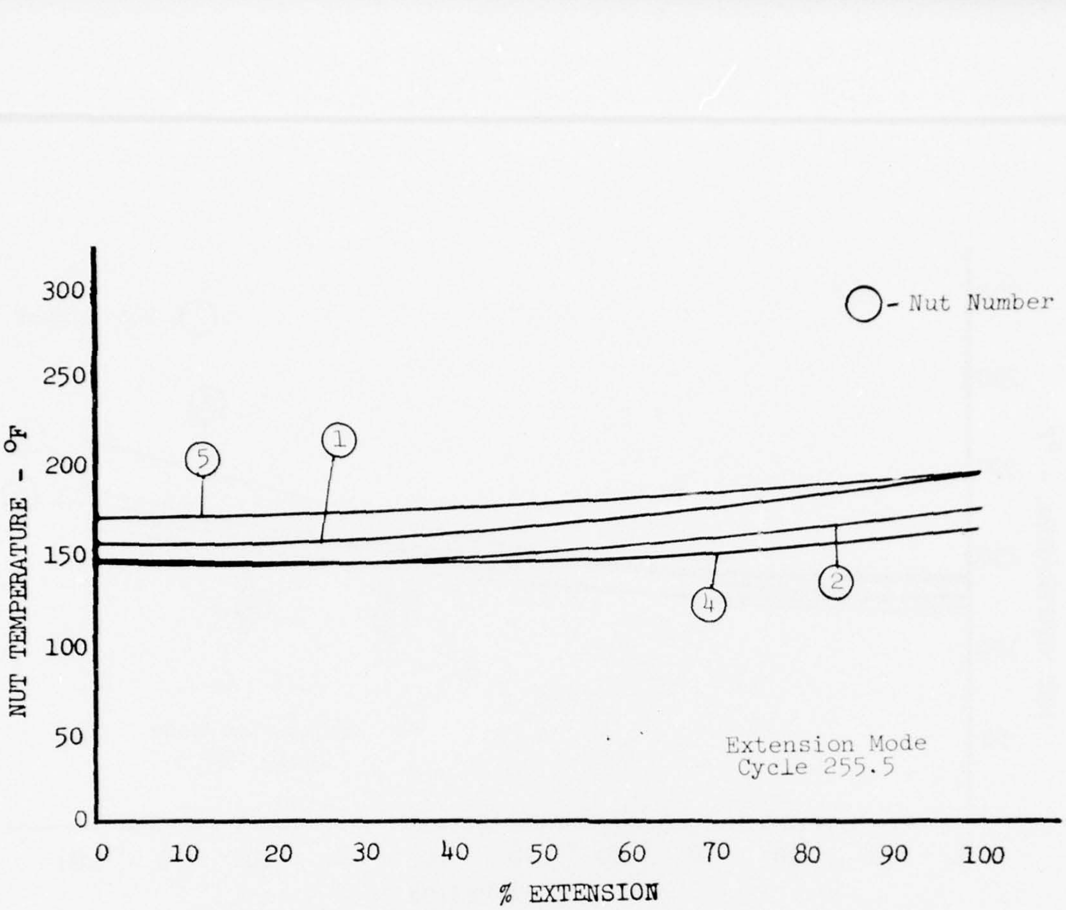


Figure 33. Nut Temperatures vs Blade Position.

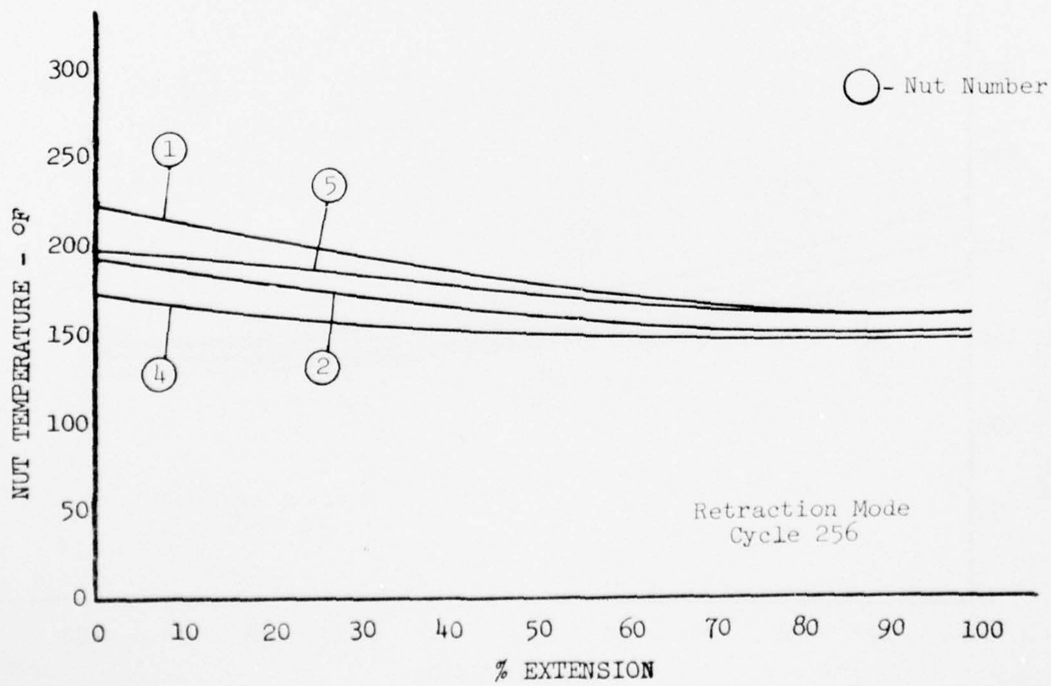


Figure 34. Nut Temperatures vs Blade Position.

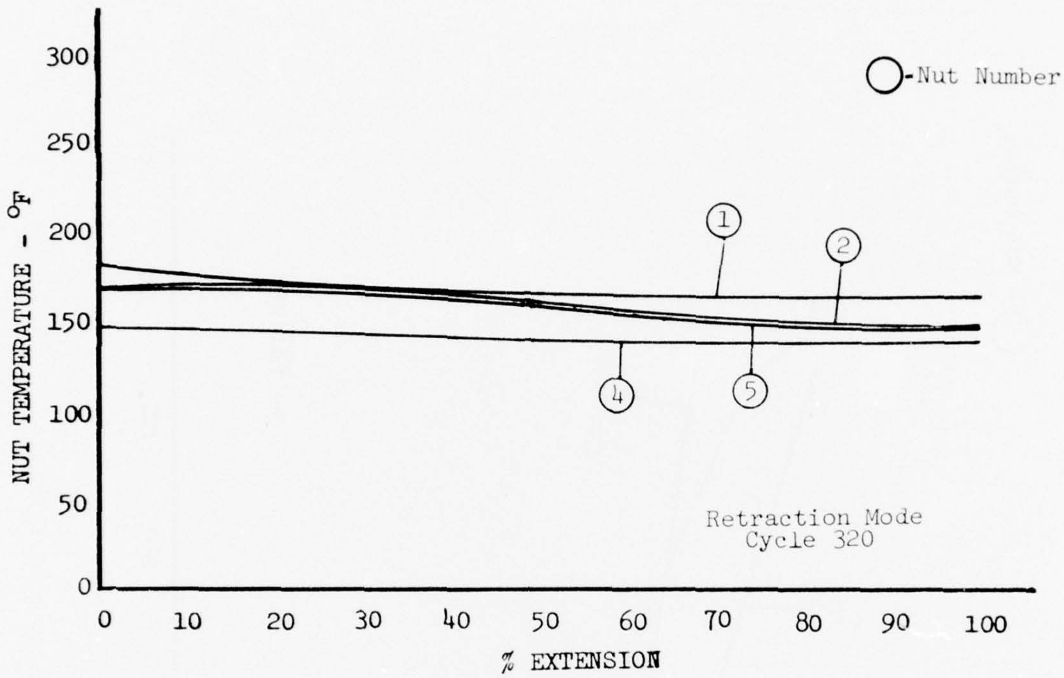


Figure 35. Nut Temperatures vs Blade Position.

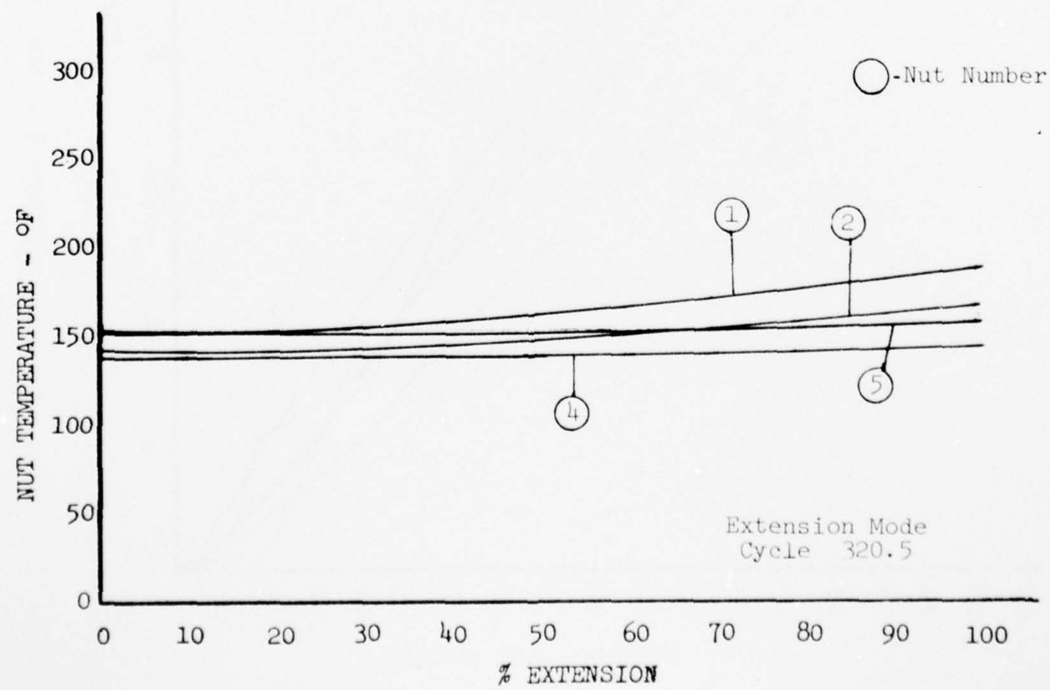


Figure 36. Nut Temperatures vs Blade Position.

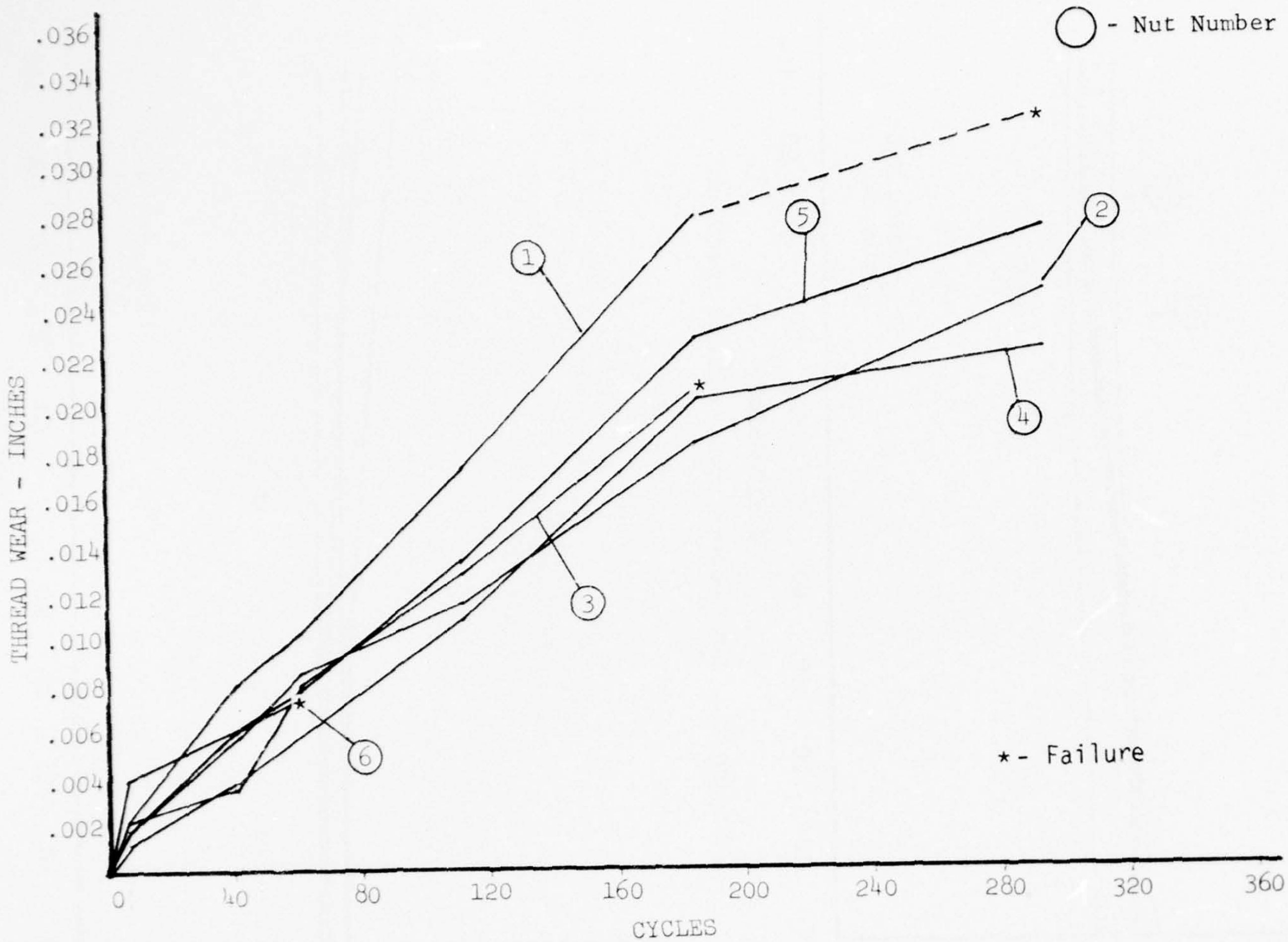


Figure 37. Thread Wear vs Load Cycles.

Test Data Analysis

The average coefficient of friction between the sliding surfaces is calculated as follows

$$\mu = \pm \frac{\frac{2T}{d_mF} - \tan\psi}{\sec\alpha \left(1 + \frac{2T}{d_mF}\tan\psi\right)}$$

where the plus sign applies to the retract portion of the cycle, the minus sign is for the extend portion, and

T = total drive torque required to overcome the thread friction and counteract the centrifugal load
F = centrifugal load
d_m = thread pitch diameter
ψ = thread helix angle
α = flank angle of the contacting surface

For the TRAC design, the above relationship reduces to

$$\mu = \pm \frac{1.17075 \frac{T}{F} - .12422}{1.03290 + .15022 \frac{T}{F}}$$

The calculated values of the friction coefficient for the measured values of drive torque and centrifugal load are shown in Figures 38 through 41.

The screw efficiency is directly related to the coefficient of friction. This relationship is as follows

$$\eta = \frac{1 - \mu \tan\psi \sec\alpha}{1 + \mu \cot\psi \sec\alpha}$$

for the TRAC configuration, $\psi = 7.08^\circ$, $\alpha = 14.5^\circ$, therefore

$$\eta = \frac{1 - .12831\,\mu}{1 + 8.3150\,\mu}$$

This relationship is plotted in Figure 42.

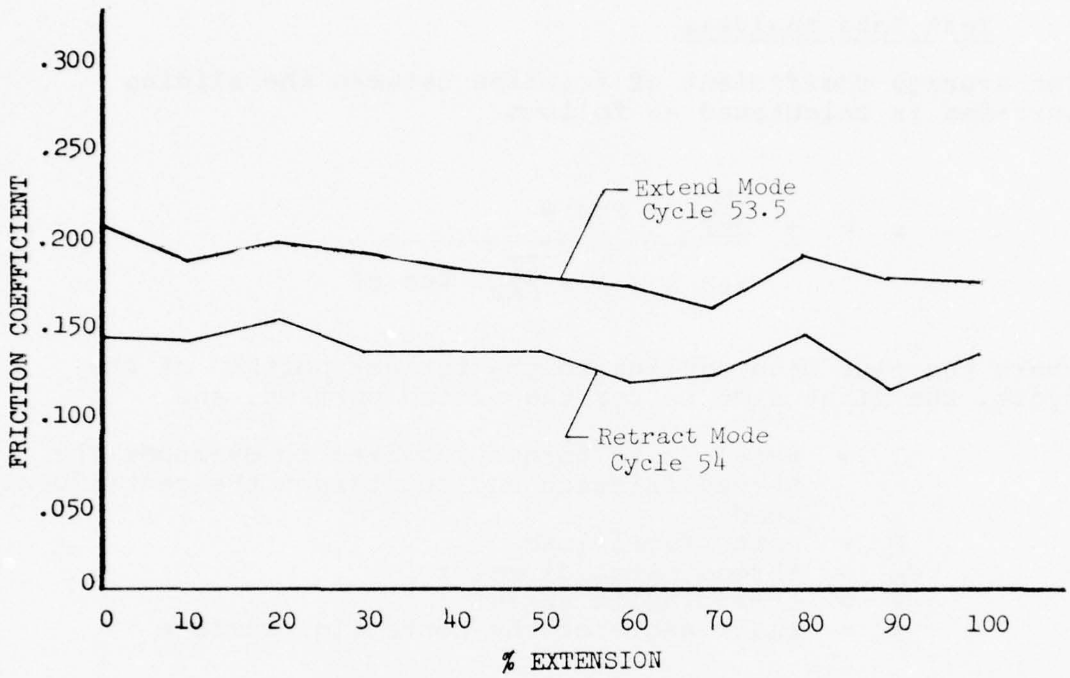


Figure 38. Friction Coefficient vs Blade Position.

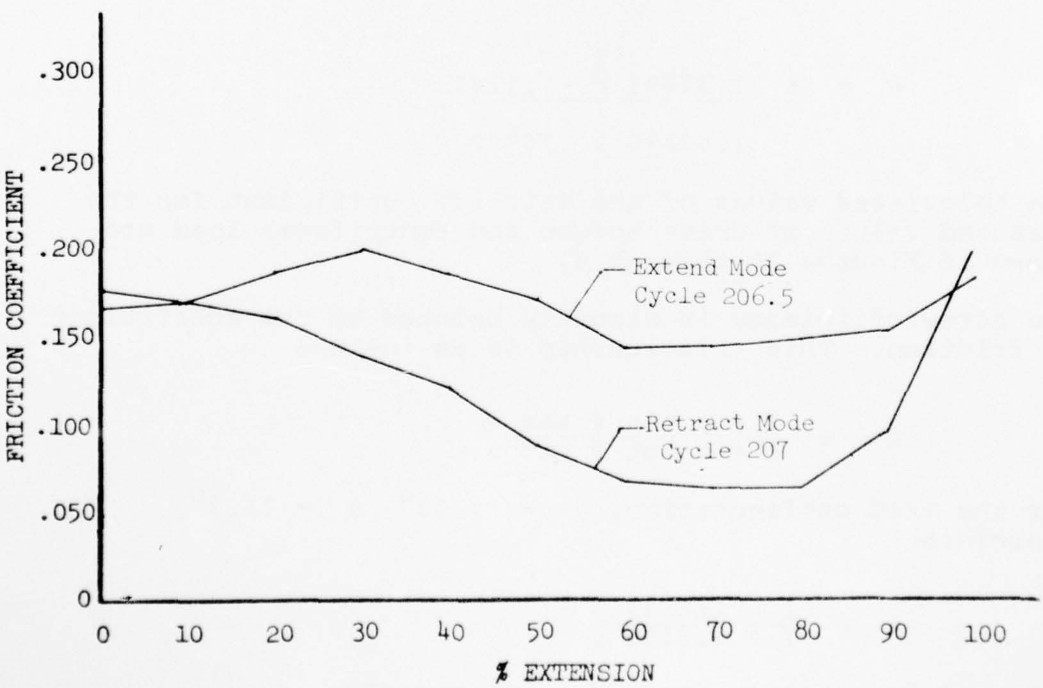


Figure 39. Friction Coefficient vs Blade Position.

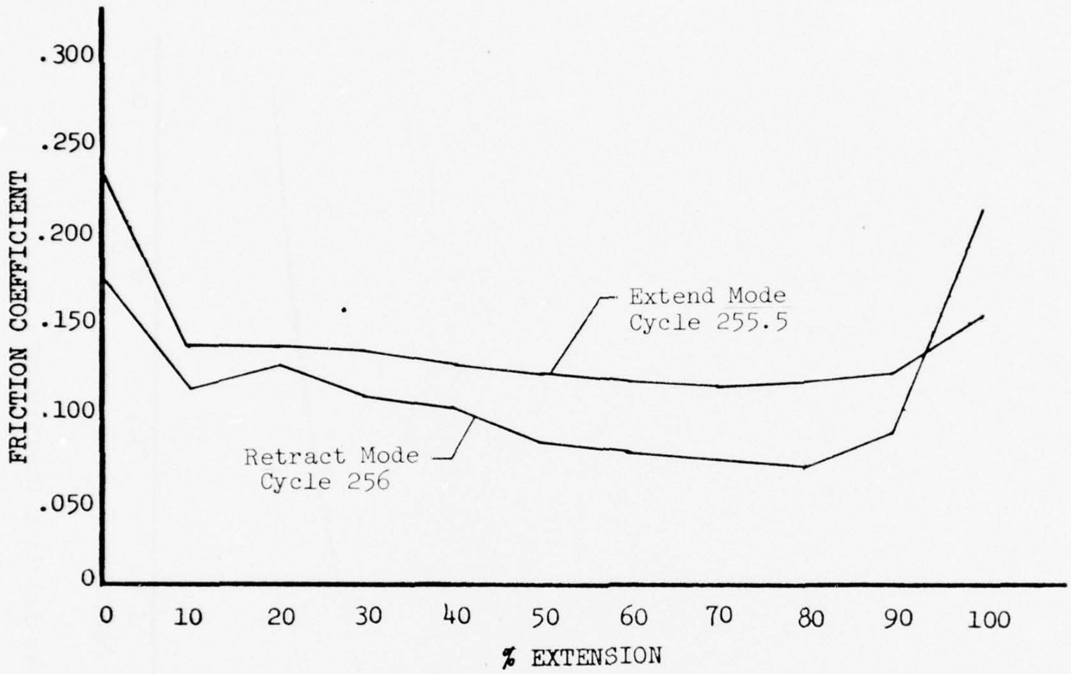


Figure 40. Friction Coefficient vs Blade Position.

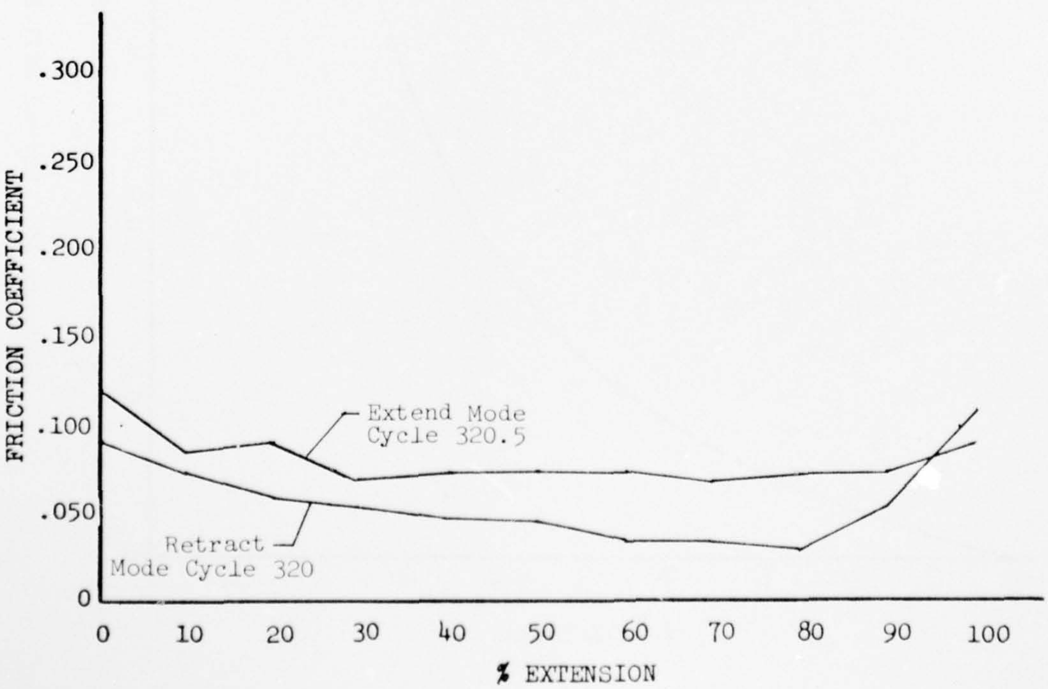


Figure 41. Friction Coefficient vs Blade Position.

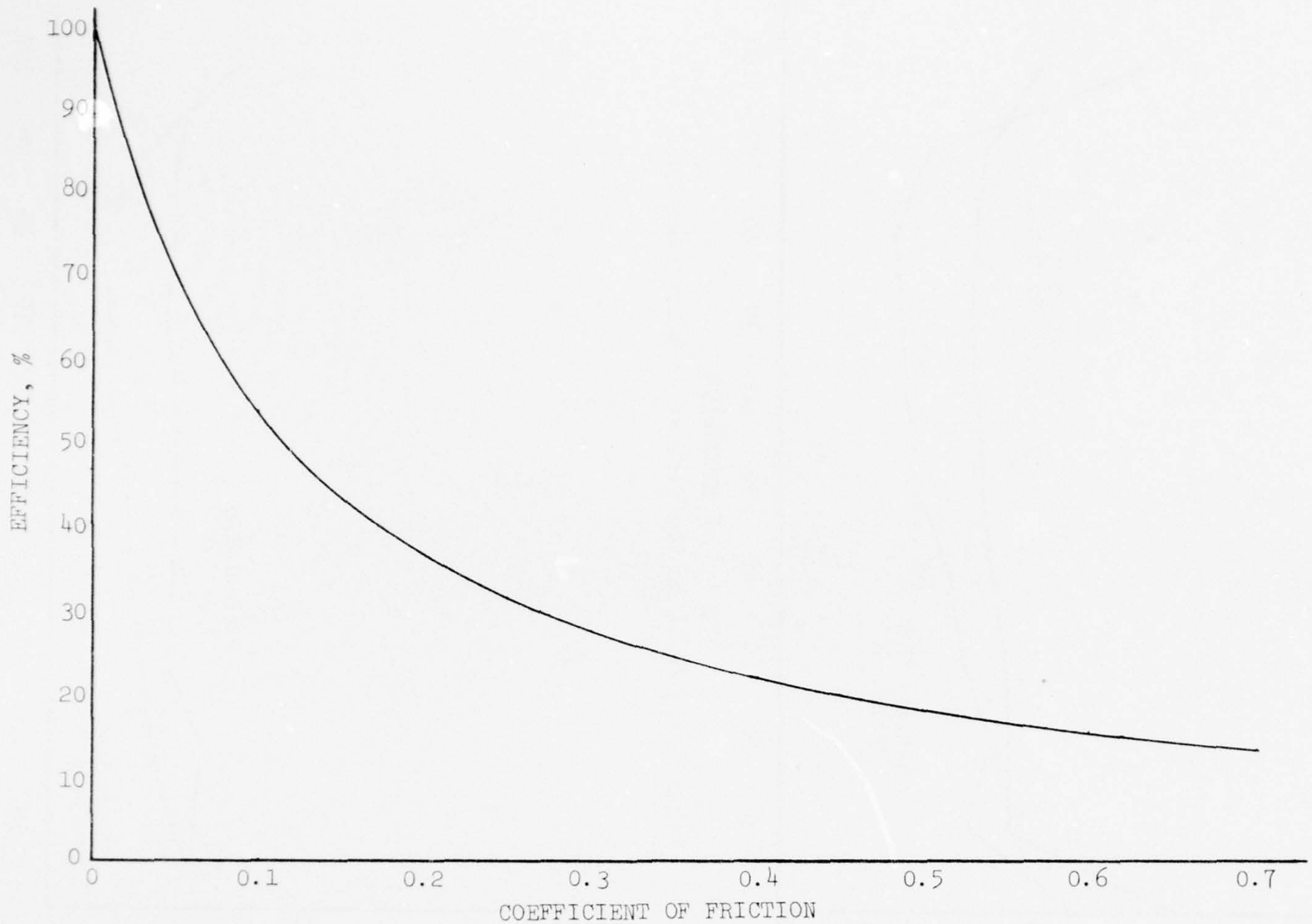


Figure 42. Thread Efficiency vs Friction Coefficient.

Discussion of Test Results

Beryllium-Copper Nuts

A number of conclusions became evident as a result of the tests on the combination of the chrome-plated jackscrew with the beryllium-copper nuts:

- The chrome-plated stainless-steel screw requires a polishing or burnishing operation after plating to remove concentrations of plating, which tends to accumulate in the thread-root fillet.

- The end of the thread of a nut should be beveled or relieved somewhat so that the lubricating film is compressed rather than scraped off during initial thread contact.

- The evidence suggests that "PV" is not the sole design criteria for screw threads lubricated with dry film. It was found that load is more significant than velocity. For example, low load with high velocity is better than high load with low velocity even though the "PV" values might be the same.

- In a double-start screw thread design, lapping of the mating parts is mandatory so that the total load can be evenly distributed among the contacting threads. It is virtually impossible to machine a thread of this form with zero error and uniform contact.

- The Vitrolube dry-film lubricant requires a burnishing operation at high speed to polish the surface and prepare it for load cycling.

Carbon-Graphite Nuts

The following conclusions and observations resulted from the jackscrew/nut tests using the preferred materials of maraging steel and carbon graphite:

- The tests successfully demonstrated that the full-scale TRAC jackscrew/nut configuration will work with satisfactory levels of friction, temperature, and wear life.

- More than 300 full retract/extend cycles out of 400 planned were accomplished before the tests were terminated by nut thread failures. Considerably more cycles may have been achieved if an excessive amount of nut material (equivalent to several hundred cycles) had not been removed in the lapping experiment. The nut run-off incident could also have contributed to premature thread failure due to the resultant shock loads absorbed by the system.

- Measurements indicated that the operation of the system was improving with time, apparently as a result of the wearing in of the carbon-graphite nut material under the applied load. The coefficient of friction decreased steadily, the rate of wear of the nut contact surfaces decreased, the temperature rise during a cycle decreased, and the noise initially present during the extension portion of the cycle disappeared, making the operation smooth and silent in both directions.

- As shown in Figures 27 through 30, the coefficient of friction, calculated from the applied centrifugal load and the measured drive torque, decreased steadily as the number of retract/extend cycles accumulated. Throughout the test, the coefficient of friction during the extension half of the cycle appeared to be greater than during the retraction half. The reason for this has not been determined, and it is inconsistent with the temperature rise, which is greater in the retraction portion of the cycle. During the later portion of the test, the friction coefficient was greater at the start and end of the cycle than in the middle portion. This may reflect the lower screw rpm during start-up and shut-down, or may indicate that the Vitrolube dry-film lubricant on the screw was less burnished on the ends than in the middle. During the later portion of the test, the coefficients of friction for both the extension and the retraction modes were less than the value of 0.12, which corresponds to a screw mechanical efficiency of approximately 50 percent and which is the dividing value between self-locking and non-self-locking modes. The lowering of the levels of friction during the extension mode appeared to correlate with the elimination of the noise encountered earlier during the extension mode, suggesting that designing the thread pitch to maintain an efficiency level greater than 50 percent would assure quiet operation in both directions. The lowest coefficient of friction recorded during the retraction portion of the cycle was less than 0.05, which corresponds approximately to the lowest levels recorded during the ring tests previously described.

- As the number of cycles increased, the temperature rise decreased, reflecting the decrease in the coefficient of friction with time. Near the end of the test, the mean temperature rise of the nuts was less than 50°F.

- Thread wear on the nuts (Figure 40) shows that the initial high rate of wear leveled out to a fairly constant value of approximately .010 inch per 100 cycles (200 half-cycles) and finally decreased to approximately one-half this rate. It would appear that a useful wear life on the order of 1000 complete cycles could be achieved if a useful wear depth of somewhere between .050 and .100 inch were designed into the carbon-graphite nut inserts.

- No measureable wear took place on the steel screw. The Vitrolube dry-film lubricant on the screw showed no apparent deterioration during the test with the carbon graphite nuts. It became highly polished early in the test program and remained that way. The only exception to this is that on two or three occasions when there was a period of days or weeks, including wet weather, between successive runs, a slight film of rust became visible on the surface of the screw, which disappeared quickly with renewed operation. The maraging steel was not rustproofed prior to coating with Vitrolube, which is slightly porous and therefore does not, in itself, provide rust protection. Therefore, it is considered desirable to provide rust protection prior to applying the dry-film coating.

- As discussed, the shear strength of the carbon graphite was demonstrated to be adequate provided that care is taken to achieve reasonably uniform contact load distribution.

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- All thread failures that occurred were fail-safe modes; i.e., the full centrifugal load continued to be carried. Thread failures occurred initially during the inadvertent nut run-off incident; the shock loads from which may have contributed to the later thread failures, which included the disintegration of two of the five threads in nut number three and finally total failure of the threads on nut number one. The measured contact surface wear rate was highest on nut number one, which probably also contributed to its failure. The low-frequency stress and thermal fatigue cycle loading are also possible contributors to the thread failures that occurred.

CONCLUSIONS

1. The tests of full-size jackscrews and nuts demonstrated more than 300 complete retraction/extension cycles at full design centrifugal load and rotational speed. If the lapping experiment had not removed a substantial amount of surface material and if the nut run-off incident had not occurred, it is probable that a greater number of cycles would have been realized.

2. The preferred material combination is high-strength maraging steel for the jackscrew and carbon graphite for the nut threads, with Vitrolube dry-film lubricant applied to the screw threads.

3. The cycling operation was completely smooth and quiet in the retraction mode. Initially, a squealing noise, apparently an alternating stick-slip phenomenon, occurred during the extension half of the cycle, but this noise disappeared after the coefficient of friction dropped to the point where the mechanical efficiency exceeded 50 percent.

4. Up to the point of failure, operation of the system seemed to improve with time, with the coefficient of friction, the temperature rise, and the nut surface wear rate all decreasing as the test progressed.

5. The wear rate on the carbon-graphite nut threads was satisfactory considering that the thread inserts are replaceable parts. It is expected that a wear life on the order of 1000 complete cycles can be achieved in a production design. There was no wear on the screw threads, and no apparent deterioration of the Vitrolube dry-film lubricant on the screw threads.

6. The temperature rise of the nuts due to frictional heating was not a problem during testing.

7. Failures of the carbon-graphite nut threads due to the inadvertent nut run-off incident and subsequent thread deterioration on two other nuts, were all fail safe, with the applied centrifugal load continuing to be carried by the remaining nuts. The inherent benefits of the multiple-nut design and the design shear strength of the carbon graphite were confirmed.

8. The beryllium-copper nut set tested in conjunction with the chrome-plated stainless-steel screw was not successfully demonstrated to be functionally suitable. The parts, which had not been lapped together prior to testing, showed evidence of mismatching: the Vitrolube lubricant was scraped off, and galling of the contact surfaces occurred at relatively low load levels. Based on the ring tests and other evidence, it is believed that this combination could be made to work properly, but this was not demonstrated.

9. The ring-test material program proved to be a useful method for simulating screw and nut contact conditions for comparative evaluation. The preferred material combination of the ring tests is the same one that worked satisfactorily in the full-scale tests.

10. This program demonstrated that a full-scale telescoping blade system, utilizing a jackscrew/nut retraction mechanism, is feasible. The retraction/extension cycling under simulated blade loads was accomplished with notable success. However, further development is desirable (1) to extend the wear life of the nut thread and (2) to demonstrate the successful interface between the bearing blocks, the torque tube, and the blade spar under simulated steady and vibratory flight loads.

RECOMMENDATIONS

A full-scale production design configuration should consist of a maraging-steel jackscrew running against a replaceable, carbon-graphite thread insert in a steel or beryllium-copper nut body.

A sufficient number of independently-loaded nut segments should be provided. Despite the fact that the thread fractures that occurred were all of the fail-safe variety, a thread of beryllium copper should be incorporated in the design of each nut that, in the event of breakage or excessive wear on the primary carbon-graphite threads, would carry the full centrifugal load.

The carbon-graphite threads should be lapped to the mating jackscrew threads to assure that loads are shared equally among the double threads. This lapping should proceed in a well-controlled sequence from low loads to high loads to preclude pitting and assure that the carbon thread thickness is not reduced to a point where its strength and wear life is adversely affected.

Some effort should be expended to determine the fatigue strength of the carbon-graphite material loaded in shear, and cognizance should be taken of the variable load spectrum in computing expected service lives and replacement intervals.

Vitrolube dry-film lubricant should be used on all metal contacting surfaces. Prior to use, the Vitrolube coating should be burnished or otherwise broken in at high speed to reduce the friction coefficient and increase thread efficiency.

For the minimum noise level, thread efficiency should be kept above 50 percent.

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